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Reference Book Sheets—As a supplement to the article on preferred numbers, we present two tables for the engineer's data book Page 431

Stampings—J. K. Olsen continues his informative series on designing for metal stampings. This time he gives design details on dowel embossings for quick and accurate assembly Page 412

Coming issues

The leading article for November will analyze and classify the results of a nationwide survey on product development conducted by the editors of Product Engineering. See page 418 of this issue for further details

Professor Hall will conclude his authoritative analysis of "Inertia Forces in Cam Mechanisms"

How patent rights are sold and rented will be explained by H. A. Toolmin, Jr.

PRODUCT ENGINEERING

Devoted to the Interests of Engineers and Executives Responsible for Product Design and Development

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In next month's issue,

PRODUCT ENGINEERING'S

Report on Current Activity in Product Development

How active is product engineering work right now, compared to previous years?

What is actually being done in the redesign of old products and development of new products?

What types of materials and unit parts are receiving major consideration?

Toward what objectives are the engineers aiming? . . . better appearance? . . . reduction of maintenance through better means of lubrication or more durable materials? . . . better performance for the user through improved electrical control, springs, gearing, or other parts?

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The extent of this activity in product development is shown in surprising terms. It is indeed a tonic, to see how hundreds of manufacturers are working to help themselves by offering better products to their customers.

This whole Report is outstanding news—an authentic forecast of the products which Industry and the Public will be buying in 1933 and 1934.

Watch for this report—in the November issue of *Product Engineering*.

PRODUCT
ENGINEERING

MCGRAW-HILL PUBLISHING CO., INC. VOLUME THREE, NUMBER TEN

OCTOBER, 1932

Inertia Forces in Cam Mechanisms

JAMES A. HALL
Professor of Mechanical Engineering, Brown University
Consulting Engineer, Brown & Sharpe Manufacturing Company

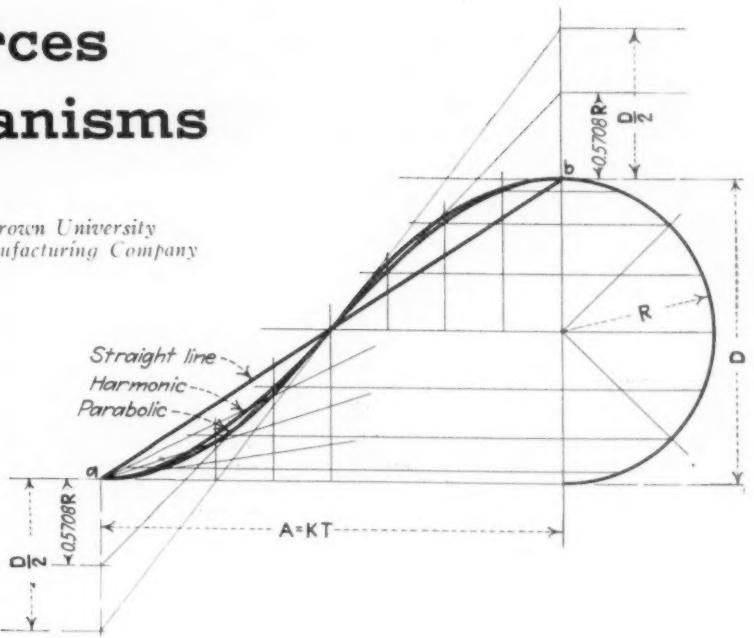


Fig. 1—Here are shown the three cam curves most commonly used. The length *A* represents the angle through which the cam rotates at constant angular velocity *K* during the time *T*, while the follower rises or falls through the distance *D*

Increased speed of operation and cutting down of the time required for non-productive movements are elements in the design of automatic machinery to which a great deal of attention is being paid at the present time. Speeding up involves greater accelerations and the resulting inertia forces may cause excessive stresses in some of the parts, and troublesome reactions on the machine frame. The parts which move with rapidly changing velocities in this type of machinery are frequently operated by cams, so the determination of accelerations and inertia forces in such mechanisms becomes of importance. Moreover, if the roller or slide is made to follow the edge of the cam by the pressure of a spring, the force necessary to eliminate play must be determined.

Where a cutting tool or a threading tap or die must advance a definite amount for each revolution of a spindle, it is essential that the cam transmit straight line motion at constant velocity. In such cases the speed of the part controlled by the cam is generally so slow that shock at the ends of the motion is not serious. The thread guide of a high speed winder travels at much higher speeds, and in spite of the fact that straight line constant velocity motion is required, a small per cent of the stroke at each end is devoted to easing off to reduce the shock of too rapid acceleration and deceleration. In other cases it is simply desired to get a part moved a certain distance in the shortest possible time and here the only problem is how much acceleration and inertia force is permissible at different positions of the stroke.

The three cam curves noted in most textbooks on mechanisms are named from the type of motion which they give the follower. These are (a) straight line or constant velocity, (b) harmonic, and (c) constant acceleration and retardation or parabolic. A fourth type, used to some extent on cylindrical cams, is made up of arcs or circles at each end of the rise to reduce the shock of starting and stopping.

Referring to Fig. 1, the length A represents the angle through which the cam rotates while the follower rises or falls through the distance D. Although A is generally given as an angle, it may be a linear distance on the surface of a cylindrical cam. If the constant angular velocity of the cam is K, and T is the time required for the given movement, the product of K and T is equal to A.

The simplest movement is one which gives a straight line from a to b. This is also most easily machined as it merely requires a constant rate of advance of the cutter as the cam blank is rotated. The velocity of the follower is constant all along the path and has a value of D/T. Theoretically the follower is subjected for an instant to infinite acceleration at the beginning of the stroke and infinite deceleration at the end. Actually, deflections of the parts, twist of shafts and similar factors, act as a cushion so there may be no trouble from inertia forces if the speeds are low.

Estimates of this shock may be made roughly by considering the force necessary to produce the required momentum in a given time or to develop the required kinetic energy in a given distance of movement. For illustration, a machine slide weighing

50 lb. is to move 2 in. in one quarter of a revolution of a cam rotating at 120 r.p.m. If straight line motion is used, the velocity of the slide will be 16 in. per sec. The momentum of the slide is given by the equation:

$$M = \frac{W}{g} V = \frac{50}{386} 16 = 2.1 \text{ lb.-sec.}$$

Where
M = Momentum in lb.-sec.
W = Weight of slide in lb.
V = Velocity in in. per sec.
g = Acceleration due to gravity in in. per sec. squared.

The value of g is given 386 in. per sec. squared instead of the more common 32.2 ft. per sec. squared. If a constant force acting during the first fortieth of the time required for the complete motion is assumed to give the follower this momentum, the amount of the force would be

$$F = \frac{M}{T} = \frac{2.1}{1/40 \times 1/8} = 672 \text{ lb.}$$

This would be the force on the machine parts and the reaction on the frame if the shocks of instantaneous pick-up can be considered to continue constantly through the time assumed.

The kinetic energy of the slide when moving at the given velocity is:

$$K.E. = \frac{1}{2} \frac{W}{g} V^2 = 16.6 \text{ in.-lb.}$$

If the yielding of the frame and the deflection of the connecting parts are such that the follower moves through the distance s equal to 0.02 inch while acquiring its velocity, and it is assumed that a constant force acts during this period, the amount of this force would be:

$$F = \frac{K.E.}{S} = \frac{16.6}{0.02} = 840 \text{ lb.}$$

If the give in the parts were elastic the force would not be constant, but would start at zero and rise to a maximum which might be double the average value. In the case given, with a deflection of 0.02 in., the maximum force might reach 1680 lb.

Another difficulty arises from the tendency of the slide to jump ahead if it is held against the cam by a spring. For instance, in the above example we might have a spring which exerts a 100 lb. force at the end of the stroke. The distance S through which the slide would overtravel before stopping and then hammer back on the cam would be:

$$S = \frac{K.E.}{F} = 0.166 \text{ in.}$$

If the assumptions are correctly chosen in this case of constant velocity motion, the inertia forces at the positively controlled end of the stroke will be excessive for ordinary machinery and the overtravel at the spring controlled end may cause much trouble. It is for reasons such as these that motions which reduce shock become important.

The harmonic motion curve is often used for the above purpose. Referring to Fig. 1, the vertical motion of the follower is given by the horizontal projection of a point moving with constant velocity along the semi-circumference drawn at point b. This motion gives a variable velocity which increases constantly from zero to a maximum at the half way position and back to zero again at b. The value of the maximum velocity is $\pi D/2T$. If this were represented by a straight line tangent to the harmonic motion curve at the center it would pass below a and above b, a distance equal to:

$$\left(\frac{\pi}{2} - 1\right) R = 0.5708 R$$

where R is the radius of the semi-circle from which the harmonic motion is constructed. The acceleration varies from a maximum at the beginning of the stroke to zero at the center and back to a maximum deceleration at the end. The maximum value is

$$\frac{D}{2}\left(\frac{\pi}{T}\right)^2$$

Applying these figures to the illustration given for straight line motion the maximum velocity would be:

$$V = \frac{\pi D}{2T} = 25.1 \text{ in. per sec.}$$

The maximum acceleration becomes:

$$a = \frac{D}{2}\left(\frac{\pi}{T}\right)^2 = 630 \text{ in. per sec. per sec.}$$

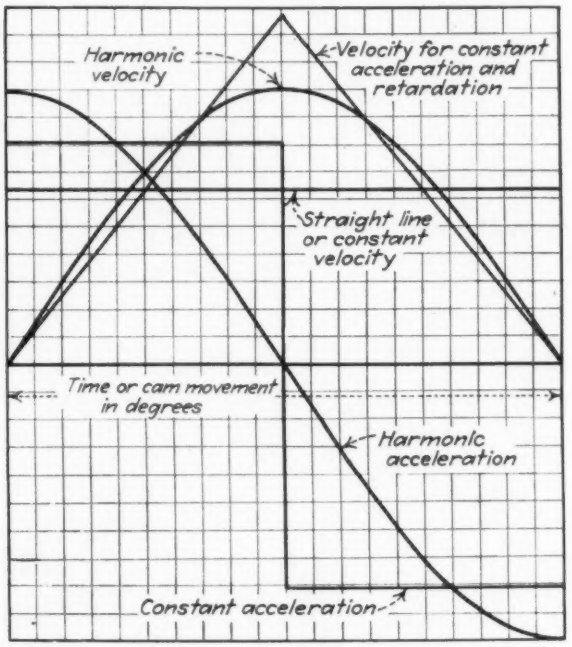


Fig. 2—Diagrams of velocity and acceleration for the three types of cam movements shown in Fig. 1, plotted against time or cam rotation as the base

The force required to give this acceleration is given by the equation:

$$F = \frac{W}{g} a = 82 \text{ lb.}$$

This is the maximum inertia force on the connecting parts and the reaction of the machine frame. It also gives the maximum spring pressure to make the slide follow the cam.

For the constant acceleration and retardation curve the cam path in Fig. 1 is made up of two parabolas which meet at the middle of the rise, which is also the point of maximum velocity. The amount of this velocity is $2D/T$ and a line tangent to the cam curve at this point would pass the starting line $D/2$ below a and cross the ending line the same amount above b. The acceleration from the starting point to the center is con-

stant as the name implies and its amount is $4D/T^2$.

Taking the conditions of the previous illustration, maximum velocity is:

$$V = \frac{2D}{T} = 32 \text{ in. per sec.}$$

The acceleration is

$$a = \frac{4D}{T^2} = 512 \text{ in. per sec. per sec.}$$

The inertia force is

$$F = \frac{W}{g} a = 66 \text{ lb.}$$

While Fig. 1 shows the position of the follower for different degrees of cam movement, an interesting comparison of the velocities and accelerations for the three types of movements is shown in Figs. 2 and 3. The former gives these values plotted against angular movement of the cam or time as a base, while Fig. 3 gives the same figures plotted against the position of the follower. In all cases the follower is moved through the same distance in the same time or angular movement of the cam.

The maximum velocity for the constant acceleration curve is twice as much as for the straight line, while the harmonic motion gives a 57 per cent increase. The maximum acceleration for the straight line motion is theoretically infinite, while that for the harmonic motion is 23.5 per cent greater than the constant acceleration value. On the time base diagram both the velocity and acceleration lines for the harmonic motion are sine curves while they come out straight lines for the constant acceleration motion. On the stroke base diagram the acceleration curve for harmonic motion is a straight line and if the velocity scale is so chosen as to make the maximum value one half the rise on the diagram, the velocity line is a semi-circle. The velocity diagram for the constant acceleration motion is made up of two parabolas meeting at the maximum point, as shown in the figure.

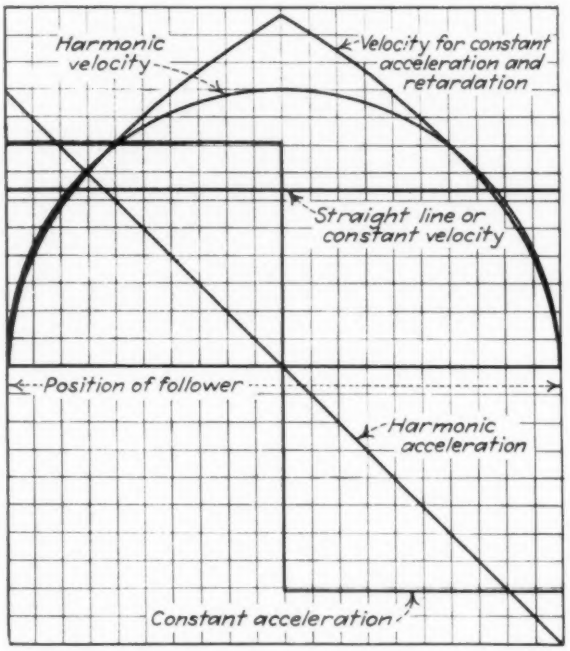


Fig. 3—Velocity and acceleration diagrams of the three cam movements plotted with respect to the position of the follower

Thus we see that the straight line motion gives the lowest velocity and theoretically infinite acceleration while the parabolic motion gives the least shock and greatest velocity. The harmonic curve falls between in that its maximum velocity is less than that of the parabolic and its acceleration is greater.

An interesting problem arises where the slide is held against the cam by a spring and the curve which will permit the most rapid drop is desired. If the curve drops too fast the cam will leave the roller and then the latter will catch up and hit with some shock. If the spring force can be considered constant the cam curve desired will be constant acceleration and retardation.

For example, take the machine slide weighing 50 lb. and acted on by a spring force of 100 lb. which is to hold it against the cam while moving through the 2-in. stroke. The rate of acceleration a will be:

$$a = \frac{F}{W/g} = \frac{386 \times 100}{50} = 772 \text{ in. per sec. per sec.}$$

If acceleration continues through the first inch of movement and retardation for the second inch, the time t required for the first half of the stroke is given by the equation:

$$S = \frac{1}{2} at^2$$

where S = the distance moved in inches in attaining maximum velocity.

$$t = \sqrt{\frac{2 \times 1}{772}} = 0.0509 \text{ seconds}$$

As the cam makes 2 revolutions per second the time t is equivalent to 36.6 deg. of movement. The maximum velocity is:

$$V = at = 39.3 \text{ in. per sec.}$$

As the time required and the distance moved through during retardation is the same as during acceleration, the total time for the movement is 0.1018 sec. and it takes 73.2 deg. of cam rotation.

While the acceleration rate is limited by the mass of the slide and the spring force, it is possible that a greater rate of retardation could be permitted safely. For instance, the acceleration might have continued through $1\frac{1}{4}$ in. of movement and retardation through $\frac{3}{4}$ in. Then the time required to reach the $1\frac{1}{4}$ in. position is:

$$t = \frac{2.5}{772} = 0.0569 \text{ sec.}$$

The velocity of the slide at this position would be:

$$V = at = 772 \times 0.0569 = 43.9 \text{ in. per sec.}$$

The retardation rate a for a distance s, equal to $\frac{3}{4}$ in., becomes:

$$a = \frac{V}{2s} = 1283 \text{ in. per sec. per sec.}$$

The time required to bring the slide to rest is:

$$t = \frac{V}{a} = 0.0342 \text{ sec.}$$

The total time for the drop under this condition is 0.0901 sec. as against 0.1018 sec. in the equal acceleration and retardation curve. It also requires 65 deg. of cam rotation as against 73.2 deg. in the first case. The inertia force during retardation

is 166 lb. however, as against 100 lb. in the first case.

There are many places where the same rate of acceleration and retardation may not be desirable, in addition to the instance given above where the motion is positively controlled in one direction and determined by a spring in the other. Perhaps a slide should be gotten out of the way as rapidly as possible at one end of its stroke to make room for another operation, or in another instance, a minimum of shock may be desired at the other end because of the necessity of accurate positioning. The valves of a gas engine must be opened an appreciable distance in a minimum time while the change at the other end of the stroke may be relatively quite gradual. Some consideration of variations in harmonic and parabolic acceleration diagrams to give these results may be of interest.

If a constant acceleration and a constant but different rate of retardation are desired, the situation can be analyzed readily on the cam development diagram by utilizing the principle that the tangent to a parabola at any point intersects the axis at the same distance beyond the apex as the projection of the point of tangency on the axis falls on the other side.

Referring to Fig. 4, the follower is to move through a distance D while the cam rotates through the angle A which equals KT. Acceleration is to go on during c per cent of the stroke and retardation during $(1—c)$ per cent. Laying off a distance cD below a and $(1—c)\ D$ above b gives the points e and f. The line ef passes from the acceleration to the retardation zone at d. The parabola having its axis at a and passing through d will be tangent to ef and so will the one starting at b and passing through d. Such parabolas will, therefore, give a smooth cam curve. The maximum velocity, as given by the slope of ef, is equal to $2D/T$, which is independent of the value of c and is therefore the same as that for equal acceleration and retardation. The time T_1 for the follower to move from a to d is cT and the rate of acceleration

$$\frac{2\ (cD)}{T_1^2} = \frac{2D}{cT^2}$$

Where c equals 0.5, the value of the acceleration will be $4D/T^2$ just as given in an earlier paragraph. If the acceleration continues through the first quarter of the stroke and retardation for the remaining three quarters, the acceleration rate comes out $8D/T^2$ and the retardation $2.67D/T^2$ or one-third as much. Another way of looking at it is that by changing from equal acceleration and retardation to increasing velocity for one quarter of the stroke and decreasing it for three quarters, the inertia force in the latter part of the stroke is decreased by one-third while that in the first part is doubled.

It is sometimes desirable to calculate the rise of a cam for each degree of cam rotation for use in accurately machining the contour. Let x = the rise of the cam curve, y = the deg. of rotation, and $a1$ = the acceleration factor, then:

$$x = \frac{1}{2}\ a_1 y^2$$

In the case given above where the slide moved one inch in 36.6 deg. of rotation, the factor a_1 can be determined by substituting 1 in. and 36.6 in. for x and y respectively in the above equation. The values of x for each degree can then be readily calculated up to the point of maximum velocity and the reverse process used coming down from the end of the stroke down to this point.

If the acceleration and velocity conditions of harmonic motion are preferred in a particular case of unequal acceleration and retardation, the cam diagram of Fig. 4 may be used again. The radii of the two circles from which the harmonic curves are developed will be $R_1 = cD$ and $R_2 = (1—c)\ D$ respectively. Distances equal to $0.5708R_1$ and $0.5708R_2$ are laid off below a and above b respectively and resulting line gh gives the slope of the maximum velocity line. The maximum velocity will come at d where this line passes from the region of acceleration to that of deceleration. A harmonic curve with a generating radius R_1 and reaching its maximum velocity at d will be tangent to the line gh and so will the corresponding curve starting from b.

The maximum velocity is independent of the value of c and equals $\pi D/2T$. The maximum value of the acceleration is $R_1\ (\pi/2T_1)^2$ and decreases to zero at d. The maximum rate of deceleration is $R_2\ (\pi/2T_1)^2$. If T_1 and T_2 are written in terms of T and c, it will be found that the relations existing between simple harmonic motion and this modified variety, with regard to acceleration and retardation, correspond to those with parabolic cam curves.

If it were considered desirable to combine the lower inertia forces of the parabolic curve at one end of the stroke with the somewhat more rapid change in velocity of the harmonic motion at the other, it can readily be done by laying off the proper amounts at each end of a development diagram similar to that of Fig. 4 and drawing in the maximum velocity line to find where acceleration ends and retardation begins.

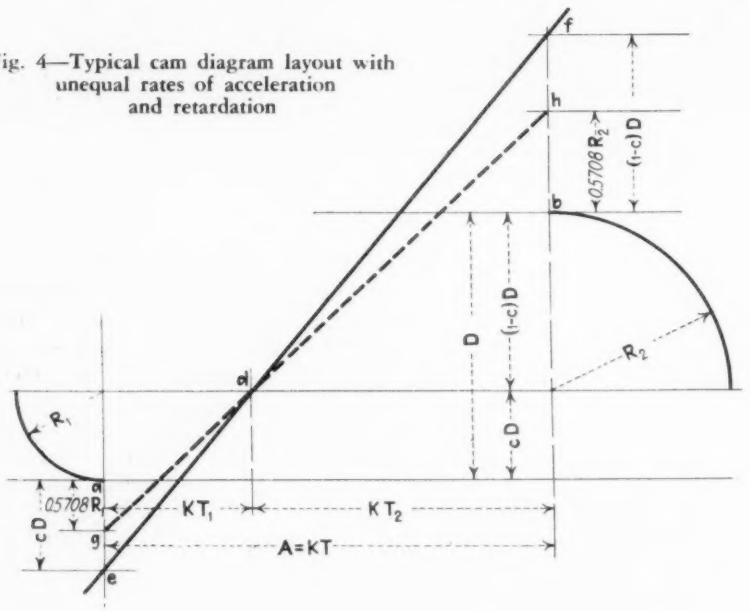
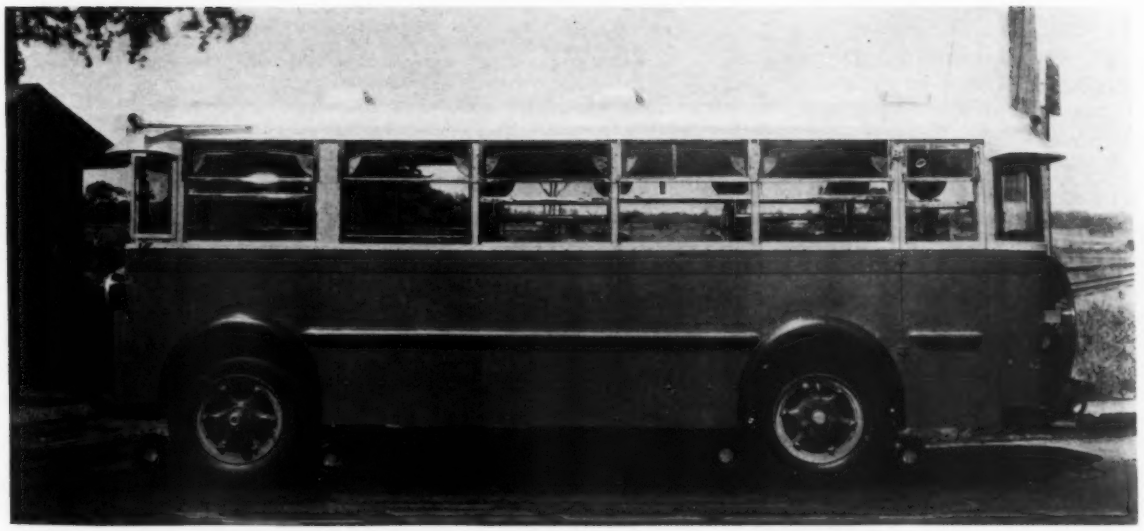


Fig. 4—Typical cam diagram layout with unequal rates of acceleration and retardation

PRODUCT DEVELOPMENTS



By means of guide wheels which can be raised or lowered at will, this Twin Coach can be operated either on rails or on the highway

▸ At Home on Rails or Rubber

Equally at home on rails or highway is the Twin Coach shown in the accompanying illustration. This unit differs from a Twin Coach standard highway bus only in that flanged guide wheels are attached to the brake spiders of a regular production job. The attachment consists of four sets of 6-in. rubber-insulated steel wheels placed about 6 in. before and after each rubber tire. In the experimental model, these are raised and lowered by means of separate screw jacks, but the designer plans to provide a control beside the driver's seat which will allow the bus to be driven on or off the rails at any point. When on the rail the guide wheels, which have flanges $\frac{1}{2}$ in. deeper than standard railroad practice, clear the rail by from $\frac{1}{4}$ to $\frac{3}{8}$ in. In case of a flat tire they will support the bus and ride it to safety. The two sets of front guide wheels steer the front rubber-tired wheels through the steering knuckles, thus obviating side sway.

The combination vehicle recently demonstrated before rail officials in Chicago is said to have attained good speed on the rails and to have negotiated switches and cross-overs successfully. The manufacturer believes the greatest usefulness of such a vehicle will be in the transportation of freight and points out that freight-carrying rail buses could be loaded at the factory and driven through the streets to the freight yard, there to be placed on track and made into trains numbering up to 50 buses. The trains would be hauled by locomotives to a distributing point where individual buses would again take to the highways and streets to deliver the shipments from door to door, without handling or loading en route.

▸ Variable-Speed Controls for Pasting Machines

The engineers of Lewellen Mfg. Co., and one of the larger Indiana paper mills, have cooperated in installing variable-speed controls on a large pasting, drying and cutting machine. The particular process is used to paste sheets of highly finished embossed board to an inner liner. The machine consists of reels from which the paper passes to the pasting machine, the pasting section, a dryer section, and the cutting section in which the finished sheets are cut into any desired length or width. It is a continuous process. The speed control problem is particularly important because of variations in the moisture of the paste, different grades of materials used, the humidity of the atmosphere, and the temperature of the dryers.

The pasting unit consists of two sections which apply paste either to the top or bottom sheets. With the first application of paste, the sheet expands, increasing in length. When two pasting operations are performed, it is therefore necessary that the second pasting rolls run at a speed somewhat faster than the first rolls. To compensate for this increase in the sheet length, a vertical variable-speed drive is used.

Just the reverse condition takes place in the dryers, however, so that it is necessary to have a second variable-speed drive so adjusted that the speed of the drying machine is somewhat less than the pasting section to

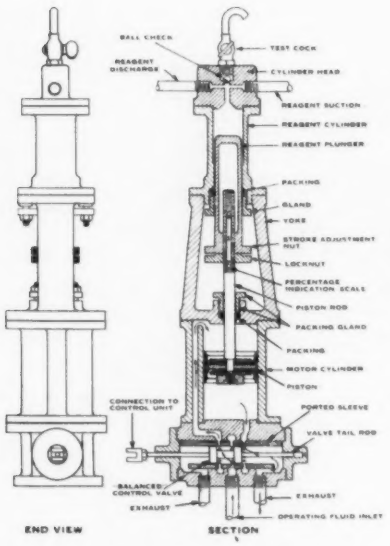
compensate for the shrinkage of the paper during the drying operation.

In addition to the controls required to maintain the proper relations between the different sections of the complete pasting machine, it is also necessary that the machine as a whole have a variable-speed control to adjust it to the various grades of paper, paste, dryer temperature, etc. For this purpose a ball-bearing variable-speed drive is used, with remote electrical pushbutton controls located at convenient points along the pasting machine. The operation of the pushbutton control does not affect the relation between the different units, it being used only to control the speed of the complete machine.

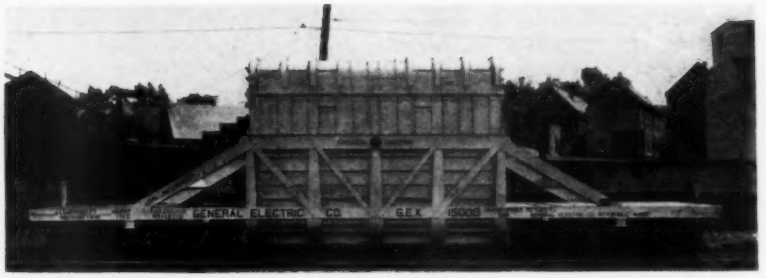
▸ Automatic Proportioning Pump

In a new automatic proportioning pump for feeding accurate percentages of chemicals or reagents in treating operations, the engineers of a manufacturer known as "Proportioneers" have developed a device which will handle ratios from a few parts per million up to normal reagent percentages with equal facility. As shown in the accompanying schematic drawing, the operation is controlled by the rate of flow of untreated fluid, the mechanism responding to the through-put of the line to which it is attached. The pump injections are frequent and in direct ratio to the quantity of fluid being treated.

The assembly consists of a hydraulically balanced control valve shuttled back and forth so as to control the number of strokes per minute taken by the small fluid motor located immediately above the shuttle valve. The fluid motor is direct-connected to a single stroking plunger pump, which creates necessary suction and discharge to pick up reagent from any receptacle, build it up to pressure, and discharge it into the untreated fluid line. The access yoke contains the reagent plunger adjusting and locking device, so that dosage may be adjusted by changing length of plunger stroke. Reagent is in contact with no part of the device except the cylinder and plunger which can be furnished in a wide range of sizes and materials to resist the actions of reagents handled. For corrosive or abrasive fluids, a diaphragm is substituted for the plunger. The "Treat-O-Unit," as it is called, can be controlled by any existing flow responsive device, such as an ordinary water meter or a rotor, or a reciprocating fluid pump.



Automatic proportioning pump for feeding chemicals in ratios ranging from a few parts per million up to normal reagent percentages



This special steel car, 60 ft. long and 10½ ft. in width, for the transportation of transformers is, except for the trucks, of welded construction. When the car, which was developed by General Electric engineers, is put into service, its tank will be filled with an inert gas to prevent the accumulation of moisture. It is estimated that over 5,000 ft. of welding were used in the assembly process

▸ Photocells in Japan

Dr. Kamo, who is at present visiting this country, reports that the engineers of one large textile mill in Japan have succeeded in substituting photo-electric cells for Jacquard cards in the production of patterns in cloth. Further data is at present not available.

▸ Speed Control by Solenoid and Rectifiers

A 1-hp. d.c. shunt motor, operating a coil winder in the East Pittsburgh works of Westinghouse Electric & Mfg. Co., operates from the 25-cycle a.c. supply with perfect speed control from standstill to full speed by dint of four small thermionic tubes and their adjuncts. Control is by foot pedal which pushes an iron core into and out of a solenoid and thus changes the power factor of the grids in two tubes.

The tubes, two 1-amp. rectifiers supplying the field coils and two 6-amp. grid glows for the armature serve to rectify the a.c. waves. The tubes, under the action of the grid, also chip off greater or less amounts of the wave, giving a variable effective d.c. voltage from a constant a.c. voltage.

▸ New Uses for Aluminum

Through the substitution of light aluminum alloys, the engineers of the Duncan Foundry & Machine Works, Inc., have saved 1,570 lb. in the weight of a new mine car, which is a saving of approximately 46 per cent of the weight of the same car in steel. Aluminum alloy structural shapes, bars and plates were used for everything except the bumpers, axle boxes and wheels. Scarcely comparable in weight saving is the adoption by Scripto Mfg. Co. of aluminum for its new double-end office pencil for two colors of writing leads. Special aluminum alloy now replaces wood in the barrel of several other new Scripto mechanical pencils.

In the field of household equipment, aluminum alloys have enabled the designers of General Electric Co. to reduce the weight of their 1,000-watt electric iron to 3 lb. By the same means, the engineers of Chicago Flexible Shaft Co. have reduced the weight of their 1,000-watt iron to 3½ lb. In the latest model gas range of the Tinnerman Stove & Range

Co., one of the features is a solid cast-aluminum burner box under the four cast aluminum open gas burners. Molded without corners to hold spilled food and dust, this box is easily cleanable and non-rusting.

Weighing but 2½ lb., yet accommodating 100 sets of standard continuous printed forms and a full-size carbon roll, the new duralumin tablet model autographic register of the Standard Register Co. will make as many as six handwritten records at once.

For similar reasons of lightness, it is reported that Frigidaire has adopted aluminum foil insulation for its newest model electric refrigerator.

▸ Roller-Bearing Pitman Roll

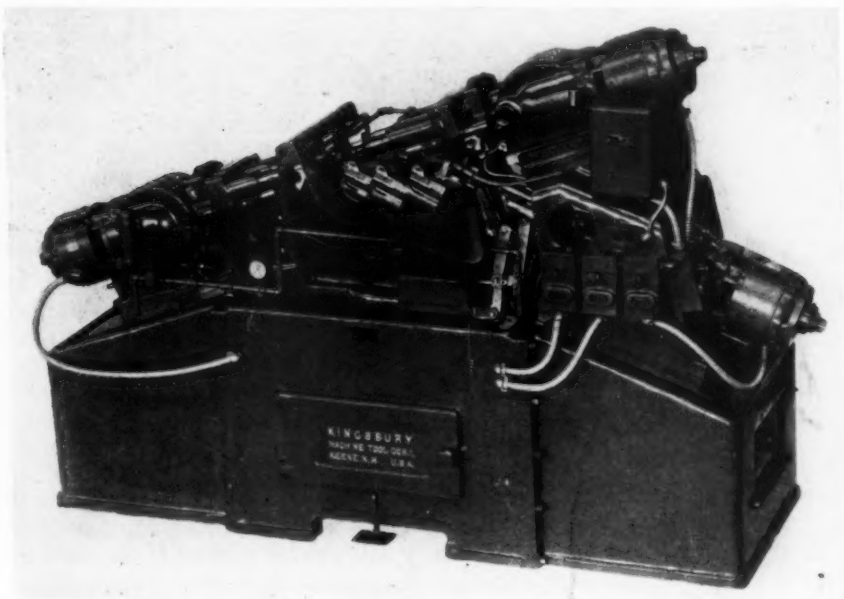
With a view to overcoming difficulties in lubricating Pitman rolls on spinning frames and twisters, the engineers of the Saco-Lowell Shops have developed a roller-bearing Pitman. The bearing is supported on a hardened steel stud which rests in recesses in the Pitman roll arm. Closely fitted steel disks are used to retain grease and exclude dust. The cam pressure is exerted on a hardened steel shell which revolves on the cylinders of the roller bearing. It is stated that use of the roller bearing prevents undue wear on the Pitman and preserves the accuracy of the bobbin build. The new roll can be applied to existing spinning frames and twisters.

▸ Welded Loudspeaker

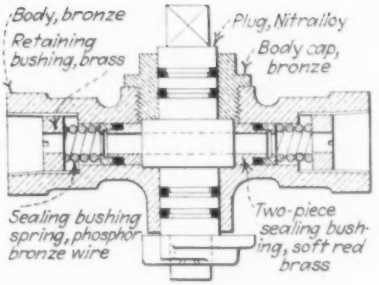
Recently reported by the Linde Air Products Co., is a theater loudspeaker completely oxwelded out of 22-gage iron. The unit, which weighs 150 lb. complete, supersedes an older model weighing 500 lb. built up mechanically out of 16-gage iron. All corners of the latter were reinforced with heavy riveted angles and channels. All parts of the new horn, including the spirals, are cut to shape with a cutting blowpipe. Jigs were designed to facilitate lining up the parts and to hold the parts while they are being welded.

Crolite, a ceramic, is being extruded and fired in V-shaped sections to replace polished agate scale bearings.

New York Central's engine 5344 of the 4-6-4 type is equipped with S K F engine-truck bearings and with Timken bearings on the driving-wheel and tender-truck journals.



In the new Kingsbury power indexing, drilling and reaming machine, a new principle is employed. The work is tilted in its nest, or fixture, permitting the drilling from opposite sides without resorting to the trunnion-type indexing fixture. Indexing is controlled by means of a foot treadle, leaving the operator's hands free at all times for loading



Yarway 1500-lb. hydraulic valve has a Nitralloy steel plug alloyed with vanadium and chromium rather than aluminum and chromium

▸ Valve with Nitralloy Plugs

Cast Nitralloy steel, alloyed with vanadium and chromium rather than aluminum and chromium, has been adopted by the engineers of the Yarnall-Waring Co. for the plug in a new hydraulic valve designed for pressures up to 1,500 lb. per sq.in. It was found that the alloy containing aluminum was difficult to cast and machined nitrided surfaces sometimes blistered.

The idea of using vanadium in place of aluminum is relatively new, although it is said to have been tested and found to have physical and corrosion-resistant properties equivalent to those of the aluminum bearing alloy. The sealing bushing made of soft red brass wears itself in against the hard Nitralloy plug and is said to give positive sealing and long life. Contrary to the drawings, expanding cup leathers are being used currently instead of ring packings. The leathers are slipped over one part of the sealing bushing, and a retaining ring is then forced against the leather ring by the spring action and an adjusting nut. It is claimed that two-piece sealing bushings give more positive sealing action and greater ease of assembly.

▸ Plastics in Three New Designs

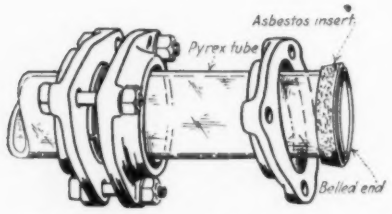
In a new office desk just marketed by the Leopold Desk Co., the designers have incorporated Bakelite molded guard inserts on the inner legs. It is reported that this effectually guards the desk against disfigurement by the banging of the swivel desk chair and prevents damage to clothing from splintered corners. And Bakelite Corp. reports that a new vacuum cleaner nozzle has been developed in England, molded out of

Bakelite instead of the customary aluminum.

In keeping with the modern trend of design among architects, The Formica Insulation Co. is producing a black plastic table top with three $\frac{3}{16}$-in. aluminum strips inlaid on the outside edges. The effect is produced by using a thin sheet of aluminum as the top lamination in the sheet and then cutting through the aluminum to expose the black phenolic resinoid between the strips. It is also possible to inlay the aluminum on top of the table in figures of straight lines, stars, shapes of animals, and so on.

▸ Couplings for Glass Pipe

The engineers of the Corning Glass Works have developed a tight and easily demountable coupling for joining glass pipe or tubing. The several parts of this coupling are shown in the accompanying drawing. To install, the tubes are swaged to a slight flare at the ends without increasing the internal diameter, but this flare is small enough so that flanges may be slipped over the end of the pipe. An asbestos insert is placed between the belled end of the tube and the flange. The flange-face gaskets are said to be



This new coupling for industrial glass pipe will hold pressures up to 50 lb. without excessive tightening of the flange bolts

of a design which insures perfect centering. The ends of the tubes are ground and grooved so as to insure tightness of the joint against pressures up to 50 lb. without excessive tightening of the flange bolts.

▸ Reporter of Idle Time and the Reasons

Idle time is the thief of profits in many a production department. Time study men are constantly on the hunt for it. The "Chronolog," invented by a former National Acme Co. superintendent and refined by the chief engineer of the Graybar Electric Co., is made by the former company, distributed by the latter, to report the amounts and causes of down time. Started by the operator's key as he arrives at his machine it serves as a time clock as well. On a paper roll the instrument prints clock time, idle minutes, productive

On a paper roll the "Chronolog" prints clock time, idle machine minutes, production units and symbols to indicate reasons for down time. The same facts appear in the windows of the instrument

minutes, units produced and symbols. The same facts appear in the windows of the instrument. A 10-hole telephone dial is rotated by the operator to record the symbol indicating the reason for down time. The symbols can be changed to suit the machine. A flashing light inside the red glass dome on top of the case functions when the machine is not producing, thus warning operator and foreman. Foremen and inspectors have individual keys which enter a slot beside the one for the operator's key. When they are at the machine a record is made of it and of how long they stay.

The chronolog can be mounted on machine brackets or on a separate pedestal. Its counting mechanism is operated by two interlocking switches, one tripped by the work piece as it leaves the machine, the other by one of the operating slides or levers of the machine. This is a safety arrangement to insure accurate recording, and the installation of these switches is the one thing about the chronolog that requires real ingenuity on the part of the installation engineer. Summary records are taken off the log sheets in the production department.

▸ Central Station Power for Suction Dredge

The suction dredge Middlesex, of the Trimount Dredging Co., engaged in deepening the Charles River, Boston, Mass., was originally operated by steam and still retains one oil-burning boiler for heating purposes and for furnishing steam to the winch engine. The cutter head and the dredge pump, however, are now operated by electric motors, using current purchased from a central power station.

With a 17-in. suction and a 15-in. discharge, the dredge pump requires a 500-hp. variable speed motor of 500 r.p.m. maximum speed. For the cutter head, which runs at a maximum speed of 25 r.p.m., a 300-hp. variable speed motor of 1,750 r.p.m. maximum speed is used, necessitating speed reducers between the motor shaft and the mainshaft upon which the cutter head is mounted. The reduction is made in two steps, the first being accomplished by a DeLaval worm gear of $13\frac{1}{4}$ to 1 ratio on the ladder of the dredge.

As the ladder is operated at various angles, from the horizontal up to an inclination of 45 deg., a forced feed lubricating system fed by a gear pump is employed which also delivers oil to a Kingsbury thrust bearing on one of the worm drives.

▸ Light-Waves to Insure Accuracy

Utilizing light waves as in indicator of contact and measuring pressure, the engineers of The Van Keuren Co. have recently developed an accurate bench micrometer. The instrument as shown has a 6-in. measuring wheel graduated in units of 0.0001 in., permitting estimated readings of 0.00001 in. A movable index is provided for quick settting to the zero position.

From a design standpoint, the most interesting feature is an optical flat which is held in contact with a 3-in. steel flat with spring pressure and is connected by a tension rod to the arm supporting the micrometer head. Thus, any movement of the light-wave interference bands shows a bending of the micrometer arm. While the instrument is operated and read as a micrometer, slight variations in size due to taper, out of roundness, or other causes, can be determined from the movement of the light-wave bands.

Fundamentals of

Standardizing Sizes and Ratings

as Related to "Preferred Numbers"—II

R. E. HELLMUND
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Westinghouse Electric & Mfg. Company

This second article, together with the tables of standard preferred numbers given on pages 431 and 432 of this issue, completes the author's analysis of this important engineering design problem. The present trend to minimize the number of sizes of units makes this discussion particularly timely. The first installment appeared in the September number

When a line of articles is developed, there are usually certain portions of the cost for tools and development that depend upon the number of sizes manufactured; in other words, the cost of such tools and engineering and development work will be doubled if the number of sizes manufactured is doubled. Let us assume then that in Fig. 10 we have this part of the development and tools cost represented by the curve MN in such a way that the ordinates at any one point give the development and tool cost for a size corresponding to such point. Thus, the development cost D of the size le would be represented by lf. It may further be assumed that the total number of pieces to be manufactured during the life of the design over a given range AB of sizes is N. If this activity N is uniformly distributed over AB, the total number of pieces over the range ax will be

$$N_x = \frac{N}{AB}\, ax \qquad (5)$$

In line with previous considerations, the dotted line ne may be assumed to be proportional to the sum of the manufacturing costs and the consequential costs of the sizes as needed, so that the ordinate $le = b$ will represent the cost C of the size le.

The development cost D of size le can be evenly distributed over all the pieces N_x which are manufactured of that size during the life of the design. Therefore, the development cost per piece will be

$$C_d = \frac{D}{N_x} = \frac{D}{N}\,\frac{AB}{ax} \qquad (6)$$

If we enter C_d in Fig. 10 in the proper scale, the shaded rectangular area may be considered as representing the aggregate development cost for the size le. The value of Equation (6), together with a number of similar factors, will be used later on in the determination of the most advantageous value for x, the percentage that a given size is larger than its preceding size.

In addition to the development and tool costs considered so far, there are usually certain other costs for engineering, development and tools which are independent of the number of sizes manufactured. Examples of this are investigations relating to the materials or methods and similar fundamental developments. Other examples are certain tool machinery which can and will be used in connection with all the sizes manufactured. Again, here may be certain tools which will have considerable wear and which will have to be replaced repeatedly during the life of the design, so that their total cost is governed by the total production rather than by the number of sizes. This also holds true when the production is so great that several sets of tools are needed regardless of the number of sizes. The costs for the tools just mentioned can be eliminated from our considerations.

The manufacturing costs previously calculated were assumed to be the basic costs and did not include any expenses for the setting up of tools prior to actual manufacture. In practice it is usually necessary in changing from the manufacture of one size to another to change certain tools and make different set-ups, frequently involving appreciable expense. It is customary to manufacture at any one time a number of pieces equivalent to the average consumption for a number of weeks or months. If such consumption or activity is N_s for the range of sizes AB, we find that the activity over the range ax for the same period of time is

$$N_{ax} = \frac{N_s}{AB}\, ax \qquad (7)$$

If the total set-up cost involved when changing from one size to another is S, we find the set-up cost per piece to be

$$C_s = \frac{S}{N_{ax}} = \frac{S}{N_s}\,\frac{AB}{ax} \qquad (8)$$

It will be noted that the derivation of Equations (7) and (8) is very similar to that of (5) and (6), except that the set-up cost is used in place of the development cost and that the ac-

tivity is for one set-up instead of for the entire life of the design.

The previous calculations regarding the set-up cost are made under the assumption that there is a reasonably large activity for each size and that such sizes are manufactured for stock. There are, however, certain cases of large machinery where practically every machine is manufactured as the order is received, and so a set-up cost is incurred for every order. If such set-up cost is independent of the size, it can be entirely eliminated from our calculations, while on the other hand, if the set-up cost is dependent upon the size, it may merely be considered as part of the manufacturing cost and treated as a part thereof. There may at times be certain intermediate cases between the two extremes just discussed necessitating calculations which form a combination of those outlined above.

The cost of keeping stock may again be divided into two parts, one or both of which may have to be considered, depending upon the case. In the case of a manufacturer dealing with large quantities, the total stock kept may be practically independent of the number of sizes manufactured. If such is the case, the cost for such stock keeping may be eliminated from our consideration. On the other hand, if the articles manufactured and stocked in large quantities by the manufacturer are marketed through a great number of small dealers, the total stock of such dealers and the expense incurred by them will be greatly influenced by the number of sizes. Let us assume that the average dealer feels that he has to have one or two pieces of each size and possibly certain stock parts for each size on hand at all times. He then incurs costs for invested capital, storage space, etc., which are proportional to the number of sizes carried. Similar costs proportional to the number of sizes may also be incurred by him because some of the stock will become obsolete before he can dispose of it, so that it either has to be scrapped or sold at a loss. For our calculations, it may be assumed that all of these costs incurred per size are represented by F. If we assume further that the activity over the rage AB for the life of a given design is N_s, the activity over the range ax will be

$$N_{fx} = \frac{N_f}{AB}\, ax \qquad (9)$$

This in turn means that the stock-keeping cost to him for each piece is

$$C_f = \frac{F}{N_{fx}} = \frac{F}{N_f}\,\frac{AB}{ax} \qquad (10)$$

If the manufacturer keeps only small stocks and the size of his stock is affected by the number of sizes the same as just discussed for the dealer, the calculations are of course exactly the same as described. Similarly, the manufacturer's stock in district warehouses may be dependent upon the number of sizes.

In cases where the user or purchaser of a article finds it necessary to keep a certain limited stock of each size, the considerations and calculations are again the same as just outlined. The same general principles apply regardless of whether such stock is kept for manufacturing purposes or whether it is kept as spares, or as spare parts for apparatus and machinery operated in the user's plant. One of the principal differences from such costs may be that the costs of spares to the user may be rather appreciable because he may, on account of the importance of continuous operation, find it necessary to keep for each size a complete spare machine in stock when he has only one or two machines of each size in operation. As in previous cases, we can write the following equation:

$$C_u = \frac{U}{N_{ux}} = \frac{U}{N_u}\,\frac{AB}{ax} \qquad (11)$$

in which C_u is the cost to the user for his stock of spare material, N_u is the activity of the range AB, and N_{ux} the activity over the range ax.

In the manufacture and marketing of most articles there are a great many clerical and routine activities, some of which may again be independent of the number of sizes, while others are dependent thereon. When an article is first marketed, certain publications and price lists have to be issued. Any part of such expense which is dependent upon the number of sizes can be readily treated as part of the development cost. Certain routine work in the works organization, such as the entering of orders, etc., might be considered as part of the set-up cost, while other clerical activities in connection with stock keeping can of course be combined with the cost of stock keeping. Then, again, there may be other routine activities of a different nature which can not be made a part of any of the costs previously treated, but which in nearly all cases can be taken care of in a very similar manner and by formulas corresponding to those given.

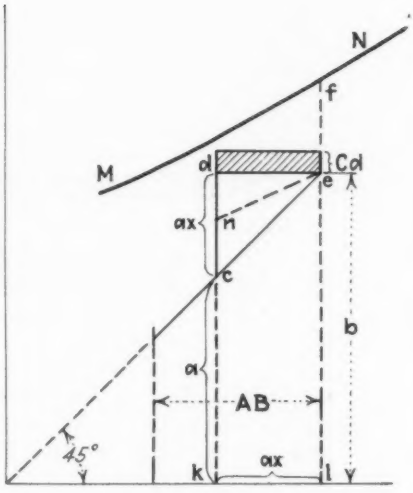


Fig. 10—Curve *MN* represents the tool and development costs that depend upon the number of sizes manufactured. Thus the development cost of the size *le* is represented by *lf*

Some, or all, of the factors discussed in the foregoing, and frequently others of similar character, may enter into that part of the first cost and of certain consequential costs of an article which depend upon the number of sizes manufactured. Some of the costs increase with decreasing number of sizes as indicated by Equations (3), (4), (4a), and (4b). While these equations were derived for the basic manufacturing costs, they are equally correct if the cost curves FG in Figs. 5 and 6 and the costs C and C_n are assumed to include the consequential costs as well as certain set-up costs. Most of the other costs discussed decrease with decreasing number of sizes. It will be noted from Equations (6), (8), (10), and (11) that the equations for all these costs are similar in character. The sum C_a of these costs as indicated by these equations is

$$C_a = \frac{AB}{ax}\left(\frac{D}{N} + \frac{S}{N_s} + \frac{F}{N_f} + \frac{U}{N_u} + \ldots\right) \qquad (12)$$

Referring to Fig. 10, we find that:

$$a + ax = b \qquad \text{or} \qquad a = \frac{b}{1 + x} \qquad (13)$$

By combining Equations (12) and (13) we find:

$$C_a = \frac{AB}{b}\,\frac{1 + x}{x}\left(\frac{D}{N} + \frac{S}{N_s} + \frac{F}{N_f} + \frac{U}{N_u} + \ldots\right) \qquad (14)$$

If we introduce a constant,

$$K_2 = \frac{1}{C}\,\frac{AB}{b}\left(\frac{D}{N}+\frac{S}{N_s}+\frac{F}{N_f}+\frac{U}{N_u}+\dots\right) \quad (15)$$

we have

$$C_a = C\,K_2\,\frac{1+x}{x} \quad (16)$$

The total cost C_t per piece, including such items as development, set-up costs, etc., then is:

$$C_t = C + C_a = C\left(1 + K_3\,\frac{1+x}{x}\right) \quad (17)$$

and if we introduce the value for C as given by Equations (3) and (4b), we have

$$C_t = C_n\left(1 + K_1\frac{x}{2+x\,(2-K_1)}\right)\left(1 + K_3\,\frac{1+x}{x}\right) \quad (18)$$

or if we wish to get the ratio R of the total cost C_t to the cost C_n, we have

$$R = \frac{C_t}{C_n} = \left(1 + K_1\frac{x}{2+x\,(2-K_1)}\right)\left(1 + K_3\,\frac{1+x}{x}\right) \quad (19)$$

We thus have a fairly simple and complete formula from which it is possible to calculate the total first cost for various values of x, assuming, of course, that there are at least approximate basic data available from which K_1 and K_3 can be determined. It is also possible to determine the value for x at which the cost is a minimum, as shown in the following

$$x^2\,(K_1 + 2K_1K_3 - 2K_3) - x\,(4K_3 - 2K_1K_3) - 2K_3 = 0 \quad (20)$$

If $a = (K_1 + 2K_1K_3 - 2K_3)$

$b = -(4K_3 - 2K_1K_3)$

$c = -2K_3$

$$x = -\frac{b + \sqrt{b^2 - 4ac}}{2a} \quad (21)$$

However, since it is usually necessary to consider various factors in addition to the first cost, it is as a rule advisable to calculate from equation 19 a complete curve, as has been done in Fig. 11. With such a curve available for a given case, it will usually be fairly easy to judge how far it is advisable to depart from the value for x at which the minimum cost is obtained, for the purpose of taking other factors into account.

Consideration of utility, serviceability, and performance has been kept until now for discussion, not so much because it is less important than the previous items, but because these three items cannot be made the subject of exact calculation in many cases. Therefore, their influence often has to be determined by judgment, along with such curves as shown in Fig. 11.

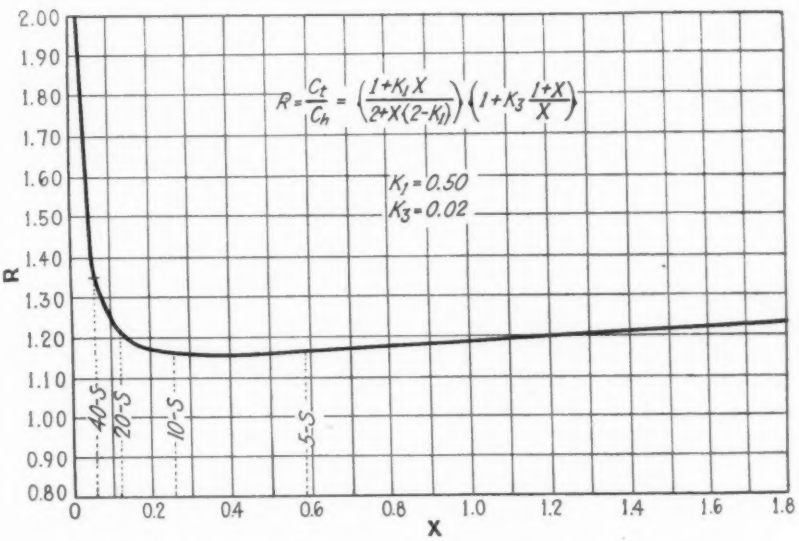


Fig. 11—Curve calculated from Equation 19, to determine for a given case the value for x for the minimum cost

Frequently, the utility of an article can be greatly increased if it completely fills a given space. For example, it may often be desired to use a kitchen range as large as possible within an available space. In view of this, it is obvious that a line of ranges covering a great many sizes will have greater general utility than a similar line having fewer sizes. Let us assume that in a case of this kind the cost curve of Fig. 11 applies, giving a minimum first cost of $x = 0.4$. It is at once evident that on account of the utility feature the 10 series having $x = 0.25$ should be chosen in preference to the 5 series with $x = 0.6$. Even the 20 series, with $x = 0.12$ and an increased first cost of 6 per cent, might be considered. It is, on the other hand, more or less evident that in so far as the 40 series would increase the first cost as much as 17.5 per cent, the choice of it would probably make the manufacturer covering such a large number of sizes noncompetitive.

In the case of machinery, the question of efficiency frequently has to be considered. Many types of machines attain their maximum efficiency at the rating for which they are designed, and if they are used for smaller loads their efficiency will be lower. Therefore, in many applications machines would have to be operated at low efficiency if only a few steps are available. The influence of efficiency can at times be taken into account analytically by capitalizing the extra expense caused by differences in efficiency during the expected life of the machine. If the curve of Fig. 11 is again assumed to cover the first cost, considerations of the efficiency are likely to lead to the choice of the 20 series in preference to the 5 or 10 series.

Another example of performance entering into the problem is that of the most commonly used type of electric motor, the induction motor. Here the power factor of the system is affected, depending upon whether there are many or few ratings available. In this case it can be shown that the influence of the power factor can be taken into account by certain additions to the first cost in case of the 5 series when its use is studied in comparison with that of the 10 series.

There are also numerous applications where the utility and performance of an article is of predominating importance as compared to the first cost. This happens very frequently with metering and measuring devices, for instance. The cost of such devices is usually very slightly affected by the rating; in other words, the cost

curve is rather flat and the value for K_1 very small. At the same time, since such measuring devices are often used in many different places, the question of keeping stock and spares makes the use of very few sizes desirable; consequently, most factors point the way toward the use of very few ratings. However, too few ratings may affect the accuracy of the readings if each of the ratings is used over too wide a range. Therefore, this item of performance may be the only one of practical importance which limits the reduction of the number of sizes.

In previous examples, the utility and performance considerations all tend toward the use of more steps than indicated by the cost curves. An example of the opposite kind is apparatus where the wear and tear is of great importance. If but few steps are available, the installed apparatus will in many cases be used below its full rating which will frequently reduce the average wear and tear. Electric contactors may be mentioned as an illustration of this. With them the wear and tear is one of the most important factors as far as utility is concerned. Therefore, if we again assume that the curve in Fig. 11 represents the first cost, it might be advisable to use the 5 series, with approximately 60 per cent steps. The use of this series increases the first cost only one per cent over the minimum, which is likely to be more than compensated for by the greatly reduced average wear and tear of the contactors in service when the 5 series is used.

Use of Larger Steps

Other instances where the utility favors the use of larger steps are those in which there is a likelihood of the load increasing as time goes on. If in such cases few steps are used, and if as a consequence many of the applications are underloaded to begin with, the necessity for installing new and larger apparatus will be greatly minimized. Another factor which is frequently of importance with regard to the utility of an article is its interchangeability with other similar articles. The chances for such interchangeability are of course the better the smaller the number of sizes, and therefore such consideration might lead to the choice of the 5 series in preference to the 10 series, with cost conditions as indicated in Fig. 11.

Many similar examples might be cited, but those given are sufficient to indicate not only that each case has to be decided upon its merits, but also that it is usually very easy to make a satisfactory choice after curves as given in Fig. 11 are available.

Even where several of the previous factors have to be considered, the proper choice of x usually does not present great difficulties. With an electric motor, for instance, the efficiency and power-factor considerations favoring smaller steps than indicated by the cost curve are usually more important than the factors favoring larger steps. On the other hand, with the control apparatus for electric motors, nearly all of the factors favor larger steps than indicated by the cost curve.

Miscellaneous Factors

There are a number of miscellaneous factors which may enter into the picture and which may influence the choice of a value for x. Let us assume, for instance, that a new type of apparatus is being developed and that again the curve in Fig. 11 gives the first cost determined from the best data available. With an entirely new type of apparatus there are usually many uncertainties regarding the design, various commercial factors, etc., all of which may bring about changes in the design sooner than contemplated. In such a case it would seem to be good judgment to start with the 5 series, with plans for later on filling in to obtain the 10 series after experience has been acquired.

In some practical cases too few sizes may entail extra expenses which cannot be readily included in the cost calculations as previously given. If, for instance, shafts of various sizes are machined from round stock, the cost of machining many of the shaft sizes will be increased if the number of stock sizes for the raw material are decreased. If Fig. 11 is assumed to apply to a cost study of this kind and the extra cost of machining has not been included in the calculation, it is at once evident that the 20 series is likely to be the best choice.

Up to this point no reference has been made to anything but straight economic features, regardless of whether the manufacturer, dealer or user carries the expense, a principle which is fundamentally correct because in the last analysis the user has to pay all expenses necessary for serving his purposes. Nevertheless, the manufacturer cannot completely ignore his own commercial situation. It is evident, for instance, that if one manufacturer has a greater number of sizes available, he is likely to secure a larger part of business than the one having fewer sizes. This in itself might be a determining factor in choosing the 20 series in preference to the 5 or 10 series in a case covered by the curve of Fig. 11.

Formula 19 used for all these considerations is based on uniform activity within the range under consideration. While it would be possible mathematically to cover the cases where the activity either increases or decreases with the size, it would lead to rather complicated expressions which probably would not find much practical application. Therefore, the more practical method of procedure is to use the formulas and curves developed for uniform activity over a certain range and to keep in mind the general tendency covering the cases for increasing and decreasing activities. If we assume again that the curve in Fig. 11 applies, but if it is known that the activity curve decreases steeply with the size, it might be advisable to use the 20 series in preference to the 5 or 10 series; while, on the other hand, if the activity curve steeply increases with the size, the 5 series should be used in preference to the 10 series.

Preferred Numbers Series

In the foregoing, repeated reference has been made to the "preferred numbers" system and it has been taken for granted in the examples cited that one of the standard series should be chosen in every case. Such a position might, however, be challenged in view of the fact that for many lines of articles the fundamental calculations on first cost and other basic considerations will lead to values of x different from those available in the standard series. Furthermore, such analysis will indicate in many cases that x should be varied for different ranges of a given line of articles because the factors such as costs, activity, etc., vary over the line. The question thus arises as to whether we should not better use in every case and for every range of a line a value of x which seems best suited and most economical according to the basic considerations.

Let us see what such a practice would lead to. In the first place, most of the factors entering into the analy-

sis, such as costs, activity, etc., are different with every manufacturer. This would naturally lead to the adoption of different values of x by different manufacturers and interfere with any national standardization. Furthermore, the various factors are likely to change with the same manufacturer as time goes on, and if he were to attempt to change around continually in order to follow a calculated value for x, he would have to change and scrap tools and equipment right along. Both this and the absence of any national standardization would lead to enormous economic wastes not included in the previous considerations. Thus it is most evident that there should by all means be a standardization by preferred numbers systems. On the other hand, it is also evident that such standardization must be sufficiently flexible to permit in all cases at least a reasonable approach to the best economic results as outlined in this paper, which is not always possible with the present standard series.

Quite a number of cases have recently come to my attention where economic considerations indicate steps of 100 to 300 per cent instead of the largest step of 60 per cent contained in the present series. There is practically no chance for bringing such cases into line with the existing system, because nobody would even seriously consider the use of two to five times as many sizes as are required to serve the existing market adequately and economically. Other cases have been observed where for some good reason or other, values of x between the 60, 25, and 12 per cent steps of the standard series were necessary or desirable. These conditions have already led to the use of series outside of the standard by individual manufacturers, and, as is natural in such cases, most of such series are influenced by personal preferences and therefore all differ, a condition which can be avoided only by recognizing the need for a more flexible system in our national standardization.

In working out any additional standards to make the systems more flexible, it is most essential that any new series be related as closely as possible to the existing standard, and the creation of entirely new series not related to the present standard should be most vigorously discouraged. As already pointed out, the reason for this is that the various economic factors entering into any one case may change as time goes on, and in some cases the changes may be so appreciable as to make a change from one series to another necessary either for part of a line or over its entire range. If this occurs, it is most important that as many of the existing sizes as possible be retained. This condition in turn can be fulfilled only if any additional series accepted as standard use numbers taken from the present series. This can be readily accomplished as follows:

Starting always with 1, going up or down,

Use every third step of the 5 series for steps of approx. 300 per cent.

Use every second step of the 5 series for steps of approx. 150 per cent.

Use every third step of the 10 series for steps of approx. 100 per cent.

Use every third step of the 20 series for steps of approx. 40 per cent.

Use every third step of the 40 series for steps of approx. 18 per cent.

With these supplements to the present standard series, which are shown in the accompanying tables (see pp. 431 and 432) together with a so-called "80 series," we can now obtain steps of approximately 3, 6, 12, 18, 25, 40, 60, 100, 150, and 300 per cent. Additional, but of course less desirable, series can also be obtained by using every fifth member of the present series. The series with the 100 per cent steps is close to one which has been proposed in a good many discussions of preferred numbers, namely,

1-2-4-8-16-32-64-128-256-508-1016,
etc.,

the only difference being that in the series proposed here and constructed from the 5 series, we have used 125, 250, 500 and 1000 for the upper figures. As previously indicated, the principal changes advocated here for constructing any additional series by using figures from the present standard series, has the great advantage that changes from one series to another can always be made with the minimum trouble and expense. If we assume, for instance, that a certain line is first standardized on approximately 100 per cent steps, such as 16, 32, 64, 125, 250, 500 and 1000, and it should happen later on that the activity and other factors change so as to require closer steps for a certain range of this line, such, for instance, that between the figures 250 and 1000, 60 per cent steps are desirable, the 500 size can be dropped and replaced by the 400 and 640 sizes without disturbing the rest of the line.

Molded Metal-Powder Products

Originally a technical offshoot of the process for making tungsten wire for electric light filaments, a process for molding small products out of powdered metal is discussed in the *Industrial Bulletin* of Arthur D. Little, Inc. The technique followed is somewhat analogous to that used in the molding of synthetic resinous molding powders such as Bakelite, although the main field for the new art will probably be the manufacture of comparatively small metal parts which would otherwise be cast or drop-forged. In addition to tungsten, many metals, including iron, chromium, nickel and copper, and also various alloys, are now being molded.

A predetermined amount of powder is placed in a steel mold and subjected to hydraulic pressure in excess of 5,000 lb. per sq.in. The formed article is then heated to cause sintering, thereby producing a metal object full of minute voids.

Essentially, the final product is a metallic sponge, varying in density from one-third to four-fifths of the theoretical density of the pure metal, depending on the size of the particle used, the pressure, the number of times it is pressed and heated, and subsequent mechanical working. The metal is not brittle.

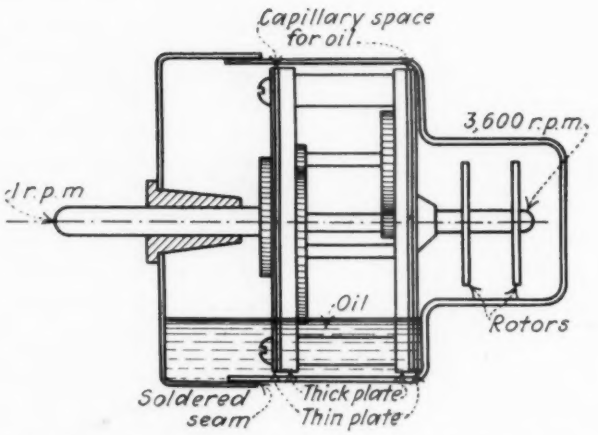
The advantages of powdered metallurgy seem to be in the control over density, composition, size and shape of the fabricated part. Alloys of constant composition and alloys which are impossible to produce by conventional methods, because they are immiscible in the liquid state, or because the difference in their melting points is too great, may be easily produced by molding powders mixed in the proper proportions. Where little strength is required, the density may be reduced to less than half without altering the appearance.

The chief technical limitation of the process is one of size. As the size of the object increases the requisite pressure increases enormously. Furthermore, there is danger of partly pulverizing the product if the mold is too deep.

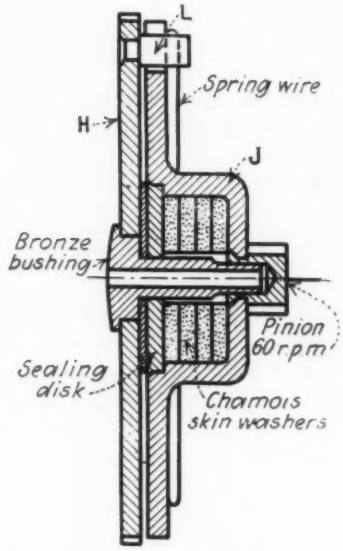
Lubrication

Examples of good modern practice in the lubrication of small bearings. These designs have the feature that no attention to lubrication is required over long periods. Several of them show applications of porous bronze bushings for long-life lubrication

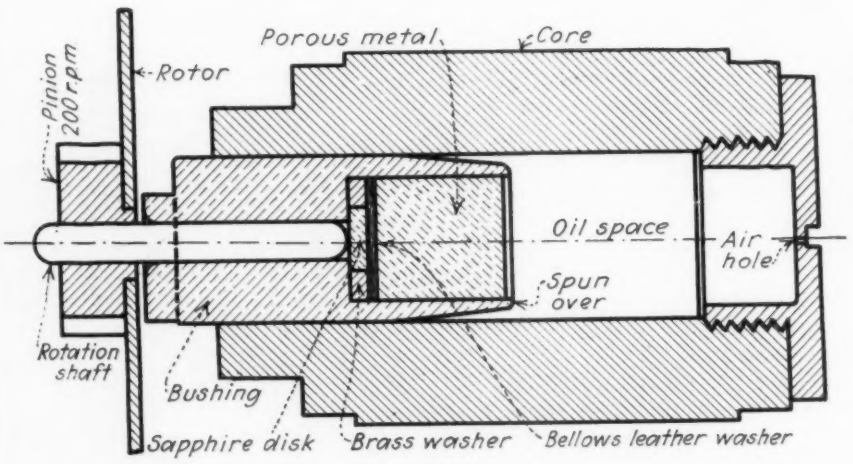
▼



This electric clock mechanism is inclosed in an oil-tight case with only a single opening for the 1-r.p.m. shaft. The bushing for this shaft projects sufficiently far into the case so that the oil level is below its inner end regardless of how the case is tilted. Oil feeds by capillary action between the plates as indicated and works out along the shaft.

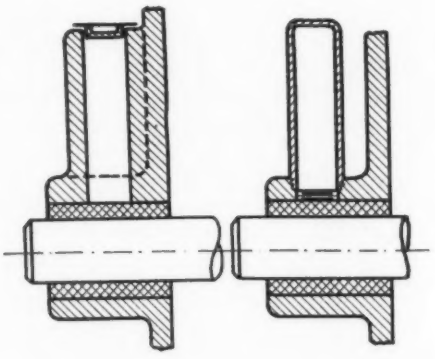


In this design, the main plate *H* of the rotor tends, because of magnetic attraction, to stay in the same place. The central bronze bushing is a press fit in the main rotor plate and turns with it. On the outside of the bushing is a floating flywheel assembly made of the cupped stamping *J*, the pinion, and the sealing disk, the latter inclosing a series of chamois-skin washers that have been soaked in oil. Oil seeps slowly through holes in the bushing, and thus to the bearing surfaces.



The rotor shaft turns in a phosphor bronze bushing pressed into the stainless steel core of the motor. The oil seeps in minute quantities through the cylindrical porous plug, through the bellows leather washer, around the floating thrust jewel or sapphire disk, and finally into the bore of the bushing in which the rotor shaft turns. The bearing clearance is held within the limits of 0.0008 to 0.0003 in.

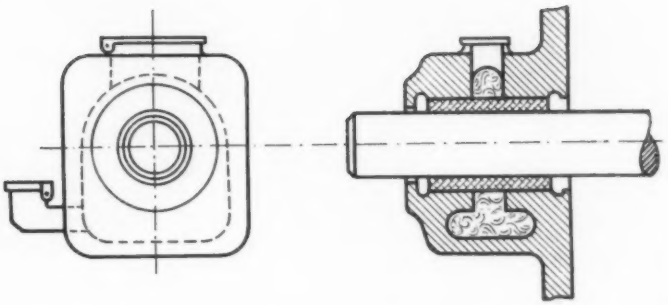
Below, to the left is an oil resrevoir cast integrally; to the right, a drawn metal shell is screwed into the hub for an oil reservoir. In both designs, lubricant is fed through the porous metal bushing. Increased bearing temperature brings additional oil to the bearing surface because of expansion and increased oil fluidity.



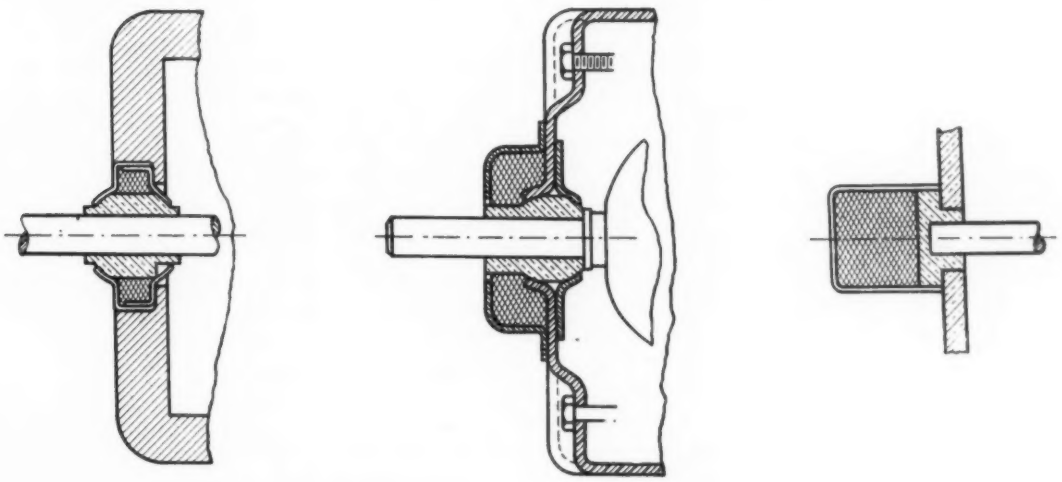
# of Small Bearings

HERBERT CHASE

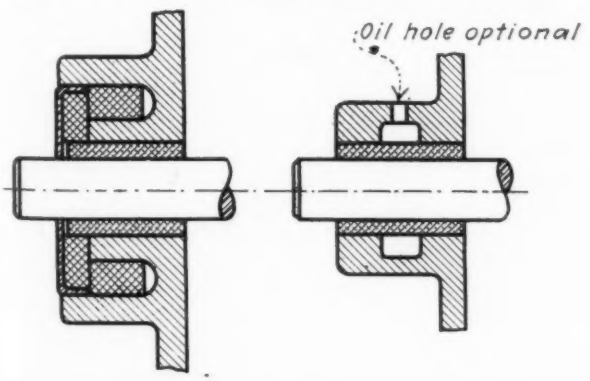
Below are shown three different designs of bearings for extremely light duty, as in clocks and meters. To the left is a self-aligning bearing having a porous bushing seated in a two-piece cadmium-plated steel shell which also holds a felt washer saturated with sufficient oil to last a year or more in ordinary service. In the center is shown a design wherein a pressed steel frame forms a spherical seat for the porous bearing. A light stamping incloses an oil-soaked felt washer that contains sufficient lubrication for the life of the motor. To the right is shown the bearing for an electric clock, the light cupped stamping that contains the oil-soaked felt being pressed over the bearing flange. In both of these designs, the lubricant in the felt is sufficient for the life of the motor.

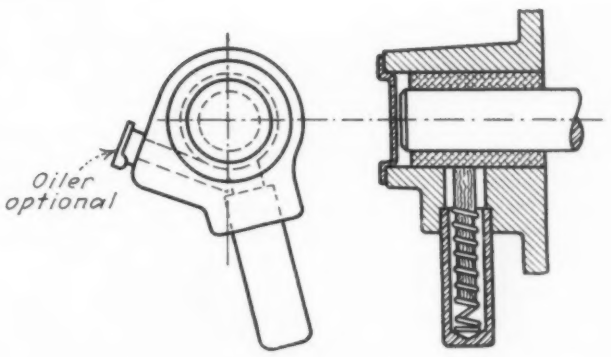


In this design, an annular groove for oil or light grease is cored in the housing of the bearing. As indicated, oil-soaked wool waste can be packed in the cored recess. The concentric grooves at each end of the porous bushing are for the purpose of catching any end leakage. A refinement would be the addition of drilled holes to lead the oil back to the cellar. The additional oiler at the bottom is optional.



A cup-shaped stamping (below) holds the oil-soaked felt in the housing, and also acts as a dust shield. Lubrication of the bearing is through the porous bushing which is die-pressed from powdered metal. The lubricant is always fed to the outer wall of the bushing, which acts as a wick. To the right is shown an optional construction wherein an annular groove in the bearing housing forms the oil reservoir.





In this wick-feed arrangement for oil or grease, capillary action feeds the lubricant to the surface of the porous bushing. For light service, sufficient lubricant is contained in the cup to last over a long period of time. The addition of the oiler shown in the end view is for the purpose of convenience so that the machine user will be more apt to give some attention to lubrication. Note the metal dust shield.

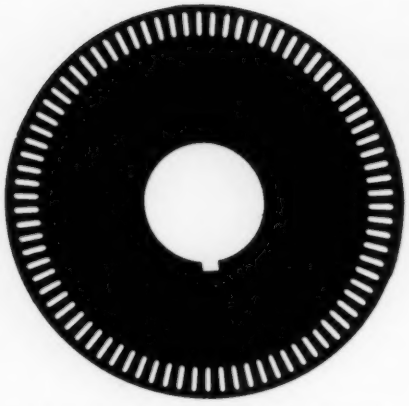


Fig. 1—Rotor punching used in general-purpose squirrel cage motor, having normal starting current and normal starting torque

Fig. 2—Deep-bar rotor punching, a type of rotor usually used in a motor having low starting current and normal starting torque

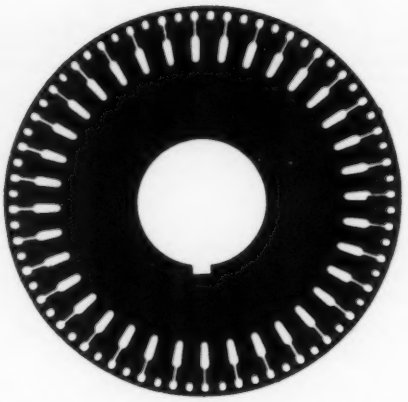


Fig. 3—Double-deck rotor punching. This type of rotor is for high starting torque and low starting current squirrel cage motors

A Comparison of Squirrel Cage Rotors

J. V. HUNT
Industrial Motor Department
Westinghouse Electric & Mfg. Company

Since the starting torque of an a.c. motor varies as the square of the percentage of applied voltage, it follows that the number of constant speed applications that a given squirrel cage motor can take care of, is dependent upon the method of starting used. For example: When using the 80 per cent voltage tap of an auto transformer the starting torque of the motor is reduced to 64 per cent of its inherent value at full voltage. Likewise, when using the 80 per cent tap, the starting current drawn from the line is also reduced to 64 per cent by means of this auto transformer type of starter. However, it should be noted that when using a resistance or reactance type of starter the starting current will be reduced only in direct proportion to the voltage reduction.

The introduction of the linestart or high reactance squirrel cage motors which are commonly thrown on the line at full voltage, and which have low starting current characteristics but develop normal starting torques, equivalent to those of the general-purpose motors, made it practical to take care of a host of applications where the starting torque would be insufficient were the voltage reduced. The primary or stator winding of this motor is such as to limit the starting current, and the secondary or rotor is likewise of special design.

Fig. 1 shows the punching used in a typical rotor of the general-purpose squirrel cage motor, which provides normal starting torque with normal starting current. Fig. 2 shows a rotor punching of the deep bar type which helps to limit the starting current and at the same provides normal starting torques.

Several years ago the introduction of the linestart double deck motor having low starting current characteristics and providing high starting torques, of from 200 to 250 per cent of full-load torque, made it possible to take care of practically any constant speed application with a squirrel cage motor. These high-torque motors will readily start plunger pumps, heavy loaded conveyors, compressors, in fact any machine requiring heavy starting torque.

Fig. 3 shows a punching used in the rotor of such a motor. It will be noted that there are provisions for two sets of rotor bars. The electrical action is such that when the motor is started most of the current flows through the outer bars which are of small cross sectional area and consequently have high resistance. After the motor gets up to speed most of the current flows through the inner bars which are of large area and have low resistance. This explains why these motors have normal slip, provide such high starting torques, and yet operate at higher efficiency than would motors which have a rotor with a single high resistance winding. The high slip, high resistance type of rotor is still used but only where frequent starting is required, as in elevator service or for such applications as punch presses, where a fly wheel is used and it is desirable for the motor to slow down somewhat allowing the fly wheel to help carry the peak load.

From the operating and maintenance standpoint, the squirrel cage motor represents the height of excellence since there are no wearing parts except the shaft and bearings. The present day rotor for squirrel cage motors is cast in one piece with the ventilating fans cast integral with the rotor bars and end rings.

Incidentally, the wound rotor a.c. motor is used where varying speed or exceedingly low starting current with high starting torque is essential.

It is of special interest to note that the average listed price of the 7½ hp., 1750 r.p.m., 440 volt linestart normal-starting-torque, normal-starting-current motor of today, with a full automatic linestarter, is about 34 per cent less than a general-purpose squirrel cage motor of similar rating with a manual reduced-voltage starter.

In addition to the great saving in first cost, the starting equipment is simplified and the full inherent starting torque of the motor can be utilized as explained above.

CLASSIFICATION OF COLORS

with reference to permanency

Prepared by **F. P. INGALLS** *Chief Chemist John W. Masury & Son*

GROUP I—*Permanent for any known length of time, and may be mixed indiscriminately without hesitation.*

Burnt Sienna
Burnt Umber
Cerulean Blue (Stannate of Cobalt)
Cobalt Blue (Alumina-Cobalt Oxide)
Cobalt Green (Zinc-Cobalt Oxide)
Chromium Oxide (Anhydrous)
Emeraude Green (Hydrated Chromium Oxide)
Graphite
Indian Red
Ivory Black
Lampblack
Titanox
Light Red (Calcined Ochre)
Venetian Red
Zinc White

•

GROUP II—*Essentially permanent under the usual conditions of exposure. They may be mixed with any number of the first group and also with each other excepting perhaps the Alizarine Lakes.*

Alizarine (or Madder) Lakes
Cadmium Yellows
Chinese Blue (or other Prussian Blues)
Manganese Violet
Ochres
Raw Sienna
Raw Umber
Terre-Verte (Ferrous Silicate)
Toluidine Red
Ultramarine Blue

•

GROUP III—*Good colors when used with intelligent discretion, but all exhibit some undesirable property or other which may lead to serious impairment of good work in time.*

Chrome Greens
Chrome Yellows
Citron Yellow (Basic Zinc Chromate)
Emerald Green (Paris Green)
English Vermilion (Mercuric Sulfides)
Lemon Yellow (Chromate of Barium)
Vandyke Brown
White Leads

•

GROUP IV—*Have no claim to permanency, although some of them are of great beauty and high saturation.*

Carmine (and all other Cochineal Lakes)
Gamboge
Geranium Lakes
Green Lakes
Indigo
Magenta Lake (Solferino)
Mauve Lake
Quercitron Lakes (Brown, Dutch and Italian Pinks)
Verdigris (Acetates of Copper)
Yellow Lakes

COLOR

Materials and Behavior in Products for Indoor Service

F. G. OSWALD
Technologist, John W. Masury & Son

Though only few generalizations about the properties of materials for indoor service are possible, the author gives some precise facts that may be applied by the product designer in the selection of color finishes

Discrimination in the selection of materials constituting finishes for products going into indoor service usually need be less exacting than for exterior service, for decomposition of finishes proceeds at a markedly slower rate in the absence of direct sunlight. Nevertheless, many special indoor conditions require individual attention, and the choice of pigments and of vehicles to carry them, may not be haphazard.

The scope of usefulness of interior finishes is extensive. Some are used for decorative effect only: certain novelties and toys depend almost solely upon color for retail buying appeal, but their transient service life makes unnecessary permanence of their color; metal curtain rods, wooden handles for electric irons, percolators, or other domestic purposes, depend as much upon color for salability, but will be used for months or years, and the customer will rightly expect color permanence.

There is now, too, a growing appreciation of the fact that colored finishes tangibly supplement the mechanical efficiency of many machines and work-bench appliances: correct color contrast between the work and the machine increases the speed and accuracy of many operations. Blue finishes, as further example, on machines in service at elevated temperatures, lessen the sense of heat to those who operate them, adding a real degree of comfort that bears measurement in improved morale and better work. When finishes are put to such uses, the designer may well take the viewpoint that the coloring materials going into them are an integral part of the apparatus, and function along with cams and levers to arrive at the utmost utility. In this sense, finishes assume their place as pure engineering problems.

Other finishes are used to provide protection in unusual industrial or consumer environments, such as in acid fumes; in high humidities; at low temperatures inducing a permanent film of condensate; or against alkaline solutions.

Thus, even a casual survey of uses for indoor finishes leads us to suspect that out of hundreds of pigment colors and dozens of film-forming liquids, some may be suited for one use and unsuited for another. And where license may be taken in certain cases, critical standards must be met in others.

Artists use a comprehensive range of pigments. Enduring work being the aim of true artists, they are much concerned with the permanence of their colors. For ready reference, manufacturers of artists' colors in oil have frequently grouped the pigments they employ according to their fastness to light and stability in other important respects. A reliable grouping is given in the accompanying table.

This grouping helps to give a more

concise view of the general properties of pigments, and may be applied as well by the technical user as by the artist. It must be borne in mind that any such list is arbitrary and those pigments that fall near the border lines between the various groups are not wholly undebatable.

Group IV includes many lake pigments not fast to light but of rare beauty and having high color saturation. *Lake* is the term applied to pigments made from natural or artificial dyestuffs, where the organic coloring matter has been precipitated as an insoluble compound and thus rendered suitable for pigment purposes. The insoluble precipitate is fixed on the *base*. The several materials used for the *base,* such as alumina, whiting, china clay, blanc fixe, etc., influence the cost of the lake, and frequently have a considerable effect upon the brilliancy and working properties of the color. Most lakes are of a semi-transparent nature, and when dispersed in vehicles produce finishes more brilliant and richer in color tone than may be effected with opaque pigments.

The attractive lake colors in Group IV find particular application in finishes for novelties and toys whose service life is short but whose color is an all-important factor in effecting sales. A large part of such finishing must be done cheaply, so that the proportion of base in the inexpensive lakes is usually high and the kind of base is determined by its cost rather than by its quality. Even starch is used for the base in the least expensive lakes.

"Coach Finishes"

Lake colors are particularly effectively presented by using them in the manner of the old coach painters. Surfaces, after being primed, filled and smoothed, are coated with opaque undergrounds of a color corresponding with the color of the lake: over the underground, one or more coats of the lake, dispersed in "japan," are applied. These coats, high in pigment, dry flat, and are followed by gloss coats of a rubbing varnish in which the same lake is dispersed, each "color and varnish" coat being smoothed by rubbing with pumice stone after drying. The successive "color and varnish" coats contain, each, proportionally less pigment. A final coat of clear finishing varnish is applied, which, in the best work, is eventually rubbed and polished. Such finishing demands artisans of highest caliber, requires weeks for completion, and naturally has been superseded by more rapid and less expensive methods. The beauty of the resultant finish, however, stands at the apex of the finishers' art. Diminishing concentrations of the semi-transparent lake pigment, over an opaque underground of carefully selected color, achieve a wealth of refracted light not obtained with fewer coats, or with opaque pigments throughout.

Compromise Methods

Compromise methods of finishing may be employed for partial attainment of the old coach finish at reduced cost. On smooth metal, a single opaque underground, serving also as the primer, followed by a coat of gloss enamel, made by dispersing the lake in a suitable clear varnish or lacquer is the simplest method and will give brilliant and attractive color results. The "depth" and color richness of such a finish exceed anything obtainable with opaque pigments, using an equivalent number of coats. It must be borne in mind that finishes utilizing these Group IV lakes are not light fast. Much better permanence is obtained with the alizarine or madder lakes, Group II, and new lake colors, which, sold under propietary names, not listed, but with somewhat better permanency than the Group IV lakes, are being introduced into the industry from time to time.

Steam turbines, automatic dry-cleaning machines having distillation apparatus built in for recovery of cleaning fluids, laundry ironing machines, and other products operating in service at elevated temperatures, present problems both as to pigment and vehicles suitable to withstand the heat. Synthetic varnishes made of glycerol-phthalate type resins are among the most valuable vehicles for such work. They not only remain physically unimpaired at temperatures up to about 450 deg. F. but also darken more slowly at this heat than do most other vehicles. Varnishes made with fish oils are somewhat more durable under continued heating than ordinary varnishes made with linseed or tung oils, but do turn amber color.

All organic pigments, whether lakes or true organic colors such as toluidine and lithol reds, change in color as temperatures increase, eventually carbonizing when the heat is excessive. They are not trustworthy in finishes that must withstand much over 200 deg. F. Metallic oxides, such as anhydrous chromium oxide, ferric oxide, zinc, antimony, or titanium oxides are well suited for such work. The cadmium sulphides and selenides offer yellows and reds fitted for high temperatures. Cerulean blue, cobalt blue, cobalt green, and ultramarine blue are also suitable.

Since blues are notably "cold" in psychological effect, they are increasingly used on apparatus of the sort here encountered, and bring real comfort to operators. Pastel tints of blue, that is, blue diluted with considerable white, are indicated rather than pure blues which, by nature, are so dark as to be almost black. Ultramarine blue with titanium oxide, or zinc, or antimony oxides, gives pleasing tints, effective except in atmospheres containing acid fumes. Acid fumes, or acid liquors, react chemically with ultramarine blue, destroying the color. In those unusual conditions envisaging acid fumes, cobalt blue or cerulean blue may be used. These blues are expensive, however, and infrequently used for industrial purposes in paints or enamels.

Unusually Permanent

Hydrated chromium oxide, known usually as viridian, is an unusual pigment in many respects. It is semi-transparent and of rather high brilliancy. Its color is markedly blue-green, and is unusually permanent with respect to acids, alkalies and light. Viridian loses its water of crystallization and changes in color when subject to high temperatures, but may be used safely for moderate heats, from 300 to 400 deg. F. The bluish character of its color classifies it also as a "cool" color. It, too, is out of the usual price range of industrial pigments, but for the occasional special use is an especially valuable material.

Anhydrous chromium oxide is a green of low color saturation, yellowish and somewhat gray. It is a green that is particularly non-tiring in its visual effect, and its stability under any set of conditions encountered industrially or in home use recommends it for many purposes, among which high temperature work is especially indicated. Its cost is not prohibitive although a little higher than most of the common pigments.

Graphite, carbon black, lampblack, aluminum powder, and metallic zinc

dust are all useful for temperature resistant coatings. Aluminum is probably more frequently employed for this type of work than any other material.

Resistant to Alkali

High speed, automatic cutting tools frequently use sal-soda solutions for cooling the tool and work. Finishes on such machine tools must be especially resistant to alkali. Phenol-formaldehyde resin varnishes, or cumarone resin varnishes resist sal-soda solutions and find herein excellent application as vehicles for undercoats and finishing enamels. The pigments used must also be non-reactive. This excludes Prussian and Chinese blues, and chrome greens, which are composites of lead chromates and Chinese blue. Other pigments and inert extenders having marked solubility in water may not be used successfully. Gypsum and Paris white should be avoided in the class of extenders, and citron yellow in the class of opaque pigments. Suitable pigments for such purpose include the earth colors, such as siennas and umbers; the carbon pigments; titanium pigments; zinc oxide; ultramarine blue; chrome yellows; white lead; chromium oxides and the expensive cobalt colors. Barytes, asbestine, and silica are satisfactory extenders. Aluminum powder is reactive with alkali and should be avoided.

Opaque Whites

Most finishing colors for consumer metal products, and many for industrial work, are tints or pastel colors, essentially mixtures of opaque white pigments with varying proportions of pigments of dominant hues, and frequently containing some black also. The sharpness of tint and its color character depend much upon the opaque white used. Zinc oxide white is unexcelled for sharpness and clearness of tint, but must be used with discretion on account of its reactivity with many vehicles, causing thickening of the paint or enamel in the can. A sufficient number of non-reactive vehicles obtain, however, to fill almost any requirement. White lead, titanium white, lithopone, zinc sulphide, and antimony oxide are the other chief opaque whites, while zirconium whites are now being introduced. Beautiful tints may, of course, be obtained with any of these whites, and each has its individual characteristics. Lead whites and antimony oxide are susceptible to color change in the presence of hydrogen sulphide or ammonium sulphide vapors. An addition of zinc oxide to antimony oxide changes the susceptibility of the combination to sulphide fumes, since a selective reaction takes place, with the formation of zinc sulphide, also white, in preference to antimony sulphide which is orange. Antimony oxide may, therefore, become practicable when mixed with zinc oxide and exposed to sulphide vapors.

In making tints, the reactivity of the pigments of dominant hue must be considered also. For example, ultramarine blue has a high reactive sulphur content which excludes it from admixture with lead pigments on account of the darkening effect of the lead sulphide that is formed. Chrome yellows, chrome greens, white lead, or red lead may not be safely used with this blue.

Good Hiding Power

Of the opaque whites, titanium dioxide and Titanox (titanium dioxide precipitated on barium sulphate) are chemically the most stable, and titanium dioxide also exhibits the highest relative opacity. This opacity permits good hiding power in enamels, paints, or lacquers at relatively low pigment concentration, and indicates this pigment for finishes in white or light tints where few coats may be employed. One coat work is frequently found possible in industrial production.

Few generalizations are possible in summarizing the properties of materials useful for colored finishes indoors. The individual properties of components of such finishes must be well known, and this information is difficult to get save through observant handling of the materials over a period of time. Unfortunately, few designing engineers are in a position to amass comprehensive, detailed facts on finishing materials for whose use they are nevertheless responsible. But the other essential factor in making the finish an efficient, properly functioning part of the machine or product, is knowledge of the machine's conditions of service. Such knowledge is within the engineer's sphere, and an intelligent union of it with the paint man's familiarity with pigments and their vehicles usually brings results, although often not without the most discriminating empirical testing.

Silver as an Engineering Material

In the past the applications of silver to mechanical design, either as a pure metal or as an alloy, have been relatively small, but recently it has rapidly come forward, especially in England and on the Continent, and is now receiving a considerable amount of attention.

According to Donald McDonald in a paper presented before the Society of Chemical Industry, fine silver is commercially available as sheet in all gages down to thin, transparent, hand-beaten foil; tubes of all diameters up to about 2 in. either welded or seamless drawn; and wire, either rolled or drawn down to 0.0005 in. Castings are not often called for, on account of the relative high price of the metal, but sound ones can be and have been made, both in the fine metal and in its alloys, up to 224 lb. in weight.

The most extensive industrial application evident so far is in the condensation and general handling of acetic acid, which is particularly corrosive at the moment of condensation. Many plants have abandoned copper condensers in favor of silver, with, as far as is known, entirely satisfactory results. Some of these are large pieces of plant weighing several hundredweights, and are of both the tubular and the coil types.

The use of silver is spreading to other allied trades, and so to the food industries. In the distilled vinegar industry the use of copper, even when tinned, has not yielded universally satisfactory results, and trouble has occurred when the vinegar made in it is employed as a pickling medium. Fine silver stills and condensers and silver-alloy taps and cocks are being generally employed.

In the manufacture of acetate rayon silk, fine silver condensers are utilized in the recovery of solvents; and the organic solutions employed require valves, cocks, and taps made from silver alloys.

Not only is the pure metal used and, to a less extent, its copper alloys, but also what is known as *doublé*, in which fine silver sheet is rolled on to copper or other base metal, thus reducing the first cost of the apparatus. Pipe lines can be made in this way by drawing a tube of the base metal over a seamless tube of silver. For the melting and casting of the caustic alkalis silver apparatus is employed.

Dowel Embossings for Assembling Stampings

BY J. K. OLSEN
Chief draftsman, Stewart-Warner Corporation

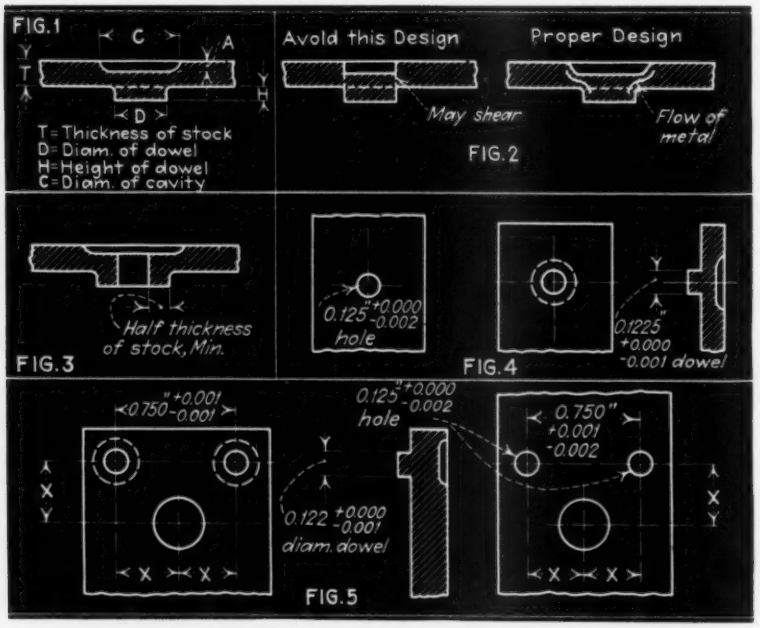


Fig. 1—The usual proportions for dowel embossings as indicated here are given in the text. Fig. 2—To the left the diameter of the cavity on the back is equal to the diameter of the dowel. To the right is indicated how the metal flows when the design of the cavity is correct. Fig. 3—When a hole is put in the dowel, the minimum wall thickness should not be less than ½ the thickness of the stock. Fig. 4—Usual design proportions for a single dowel construction. Fig. 5—All of the dimensions on the part should be given relative to the dowel

In this fourth article the author explains the use and design of dowels in stamped metal parts. The next article will deal with stamped parts involving bending or forming operations, and the manner of calculating the developed lengths

To obtain accurate alignment of parts in assembling, dowels in either flat, bent or drawn sheet metal parts are now utilized. These embossings, which are almost always round, offer both the designer and the production man a highly efficient means for accurate positioning at assembly. There are certain restrictions in designing stamped dowels, but the data given here will be found sufficient for all designs except perhaps some extremely unusual ones.

The first problem is generally the height of the dowel, since in most cases it is desirable to make it as high as possible. Ordinarily this height should not be more than ½ the thickness of the stock, although with the softer grades of stock it is permissible to make the height of the dowel ⅝ or even ¾ the thickness of the stock. The usual proportions are shown in Fig. 1 wherein:

H = from ½ to ⅝ T
C = approximately $D + \frac{1}{2} T$
A = Volume of stock required to fill the portion bounded by diameter D and height H against the equivalent volume bounded by A and the diameter C. The cavity A can be figured approximately or found by the "cut and try" method.
D = the diameter of dowel should not be made either too large or too small. It is advisable to make it not less than ½ T, especially on the thinner gages.

The reason that the diameter of the cavity on the back of the dowel proper is made larger than the diameter of the dowel is to avoid shearing through the stock as in punching a hole.

As shown in Fig. 2, if the diameter of the dowel and cavity were the same, the punch in forcing out the dowel would in most cases shear the stock and in this manner leave only a slug. Furthermore, the depth of the cavity must be the same as height of the dowel. This type of dowel should not be used. The second design shown to the right is correct, with the cavity opposite the dowel larger in diameter than the dowel. This large diameter of cavity also gives more stock to form the dowel, the metal flowing into it as indicated in the illustration. Thus the strength of the dowel is increased and shearing is avoided.

It is sometimes desirable to have a hole in the dowel portion. In designs of this kind, illustrated in Fig. 3, the stock in the wall of the hole should be at least approximately equal to the thickness of the stock and not less than ½ the stock thickness.

The most common design of stamped dowels is the single dowel located in relation to some other important feature of the design. The

single dowel arrangement may not always be the most satisfactory, but conditions may be such as to confine the construction to this type, such as shown in Fig. 4. In this design a standard size punch 0.125 in. diameter is specified for the punched hole into which dowel is to be assembled. The diameter of the dowel is made to suit, in this instance maximum 0.125 in. and minimum 0.123 in. The maximum diameter of the dowel is made 0.1225 in. and the minimum 0.1215 in. The minimum clearance will be the minimum hole 0.123 in. minus the maximum diameter of the dowel 0.1225 in. or 0.0005 in., that is 0.00025 on each side. The maximum clearance will be the maximum hole 0.125 in., minus the minimum diameter of dowel 0.1215 in. or 0.0035 in., which leaves 0.00175 in. on each side.

In the practical application of dowels, it is often an advantage to have at least two dowels located in definite relation to each other. The dowels located as shown in Fig. 5 with all dimensions on the part located from these dowels, will always provide better alignment of stamped parts in assembly. It also makes it possible to keep variations down to a minimum. The example shows a typical arrangement and design of stamped dowels. It is, however, common practice in dowel designs of this kind to stagger the dowels also as in Fig. 6, so that they can be assembled in only one way.

In dimensioning a design incorporating stamped dowels of this kind, an interesting problem of specifying the proper production limits is involved. The dowels should be dimensioned with such a set of limits that they have practical production tolerances that do not impose hardship on manufacturing and still serve the purpose of locating parts in relation to each other with the least variation.

In analyzing a stamped dowel design such as described above and wherein the sizes of the holes and dowels are known, the maximum and minimum center distance can be calculated by the following simple method.

Distance between holes, referring to Fig. 7, is as follows:

1. The maximum distance between holes equals, maximum center distance, 0.751 in., minus minimum diameter of hole, 0.123 in., (half of hole diameter on each side) or 0.628 in.
2. The minimum distance between holes equals, minimum center distance, 0.749 in., minus maximum diameter of hole, 0.125 in., (half of hole diameter on each side) or 0.624 in.
3. The maximum distance to the outside of holes equals, maximum center distance, 0.751 in., plus maximum diameter of hole, 0.125 in., (half of hole diameter on each side) or 0.876 in.
4. The minimum distance to outside of holes equals minimum center distance, 0.749 in., plus minimum diameter of hole, 0.123 in., (half of hole diam. on each side) or 0.872 in.

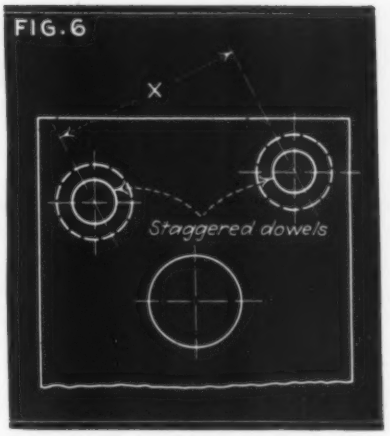


Fig. 6—To provide fool-proof assembly, it is common practice to stagger the dowels

This same procedure is followed in figuring the distances between the dowels.

5. The maximum distance between the dowels equals, maximum center distance, 0.751 in., minus minimum diameter of dowel, 0.121 in., (half of dowel diam. on each side) or 0.630 in.
6. The minimum distance between the dowels equals, minimum center distance, 0.749 in., minus maximum diameter of the dowel, 0.122 in., (half of dowel diam. on each side) or 0.627 in.
7. The maximum distance to outside of the dowels equals, maximum center distance, 0.751 in., plus maximum dowel diameter, 0.122 in., (half of dowel diam. on each side) or 0.873 in.
8. The minimum distance outside of dowels equals minimum center distance, 0.749 in., plus minimum dowel diameter, 0.121 in., (half of dowel diam. on each side) or 0.870 in.

With these eight factors known, the next step is to determine whether the dowels will enter the holes under both maximum and minimum variations, as follows:

9. The maximum inside distance between holes equals 0.628 in. and the minimum inside distance between dowels is 0.627 in. With these extreme conditions, we have a total of 0.001 in. press fit or a 0.0005 in. push fit in each hole.
10. The minimum inside distance be-

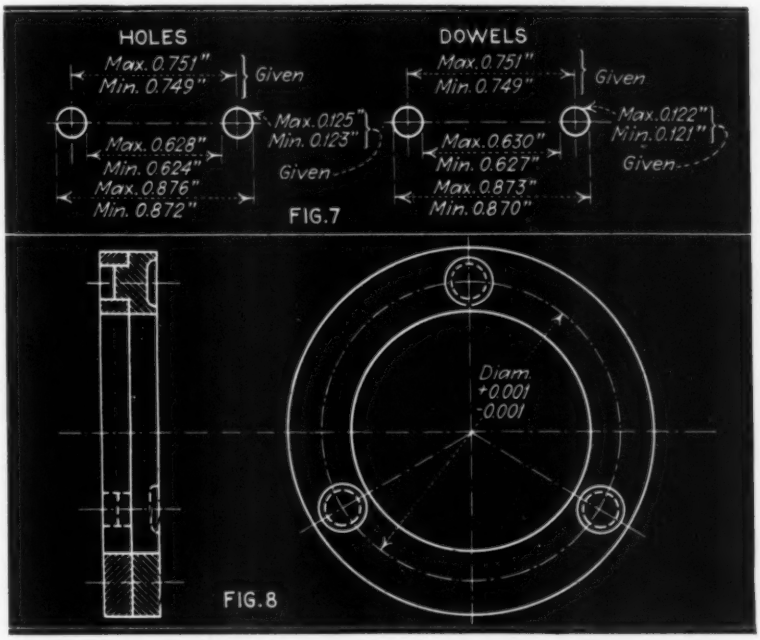


Fig. 7—By indicating the limiting dimensions as shown here, it is a simple matter to check the design to determine clearances. Fig. 8—When the dowels are put in a circle, the limits are specified on the diameter of the locating circle

tween holes equals 0.624 in. and the maximum inside distance between dowels is 0.630 in. which shows in this case that in the extreme condition there is a total of 0.006 in. clearance or 0.003 in. clearance in each hole.

11. The maximum outside distance for the holes equals 0.876 in., and the minimum outside distance for the dowels is 0.870 in., which shows that in this extreme condition there is a total of 0.006 in. clearance or 0.003 in. clearance in each hole.
12. The minimum outside distance for holes equals 0.872 in. and the maximum outside distance for dowels is 0.873 in. which shows that in this case there is in the extreme condition 0.001 total of press fit or 0.0005 in. push fit in each hole.

As shall be explained, this 0.003 in. clearance to 0.0005 in. push fit in the holes under both extreme conditions should be figures practical enough to accept for most designs.

The plus and minus 0.001 in. limits between hole centers is a rather liberal tolerance since it is only a matter of accurate punch and die work, which can be held to close limits. The chances that both the center distance of the holes and the center distance of the dowels would both be made either to the minimum distance between holes or to the maximum distance between dowels or vice versa, should be very remote.

The 0.001 in. limit on the diameter of the dowel is the closest of the limits specified. There should be no trouble holding this limit when we consider its method of manufacture, since the punching or rather swaging of the stock into a solid hole can be held extremely accurate.

As the variations specified on the hole are liberal, for most holes, this limit can be made closer. If necessary 0.001 in. instead of 0.002 in. can be given. This of course depends to a large extent on the size of the hole and the thickness of the stock, also on the importance of the alignment requirement. This dowel hole can even be made considerably closer if a shaving operation is added, but the widest possible variation should always be specified.

A good old practical production rule can be made use of in this set of figures for the dowel and dowel hole. This rule can be understood better if we ignore the possible 0.001 in. press fit at both outer and inner extreme conditions. This practical production variation rule is to take care of only the largest percentage of variations and to assume that the small percentage of variations will never occur. In other words there may never be a 100 per cent variation both ways when there are a number of variations present. It does, however, take good judgment to select the proper dividing constants.

In the above example where we have from 0.006 in. clearance to 0.001 in. press fit, we have a total of 0.007 in. variation. In this simple problem there is approximately 85 per cent chance for clearance and only approximately 15 per cent chance for push fit. In this case it is good common sense to take this chance with the 15 per cent push fit, and ordinary production will prove this satisfactory. There are many instances where this kind of chance must be taken if desired results and costs are to be satisfactory. If these limits should be produced to the full extent, a slight change in one or the other parts could be made after a check of all the component parts has been made.

Stamped dowels are also used in designs where more than two dowels are incorporated. This is especially in round parts where it is important to have two or more dowels to insure that the parts are centrally located in relation to each other. It is common practice in this type of design to use three dowels.

As can be seen in Fig. 8, instead of specifying the plus 0.001 in. minus 0.001 in. limits between the punched dowels in this arrangement, the limits are specified on the diameter of the locating circle. In this manner the computing of the extreme variations is done the same as in the previous example of two punched dowels, since in this respect it is similar to the two dowel design. When stamped dowels of this kind are used, it also happens that a third member can be incorporated to advantage.

There are different arrangements in this type of design and if thickness of stock is favorable this thickness would be beneficial in locating a third part by the use of the punched dowel in one of the parts. The design shown in Fig. 9 is a typical example of this kind which can be used in either single or multiple stamped dowel designs.

In the design shown in Fig. 9 there are three parts located by one dowel and two dowel holes. By referring to the figure it can be seen that two

(Continued on page 418)

FIG. 9

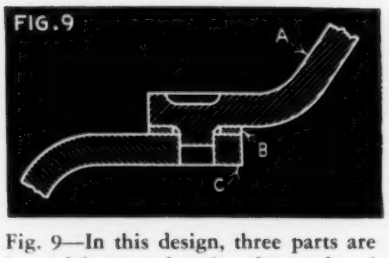


Fig. 9—In this design, three parts are located by one dowel and two dowel holes, the piece between being thin

FIG. 10

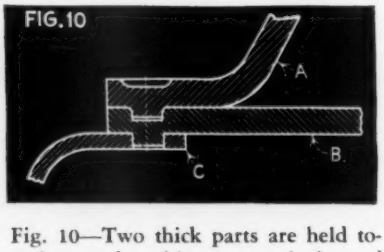


Fig. 10—Two thick parts are held together, and a thinner part is located against one of the heavier pieces

FIG. 11

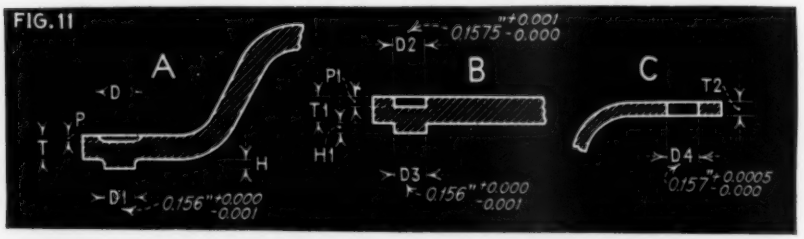


Fig. 11—The above figures indicate the correct proportioning of dowels that locate three parts in relation to each other

FIG. 12

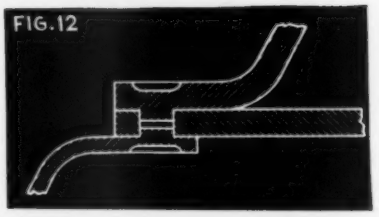
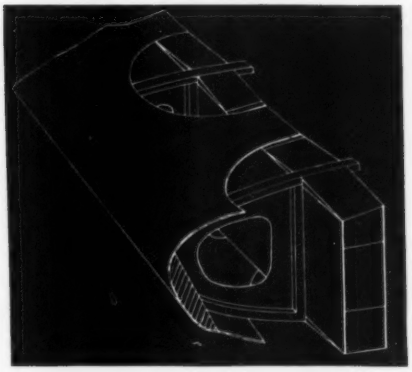
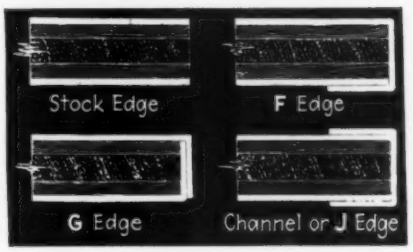


Fig. 12 — An alternate method of using two dowels and one hole

PLYWOOD
as an Engineering Material





Above—A plywood leading edge in an airplane wing presents a smooth contour and retains its shape. Below—Three methods of bending over the edge and capping metal-covered plywood to protect the interior against moisture

It is only since the introduction of water-resistant glues that plywood can be considered as an engineering material having definite physical properties such as weight, strength, elasticity and hardness. Natural wood is a non-isotropic material with the strength along the grain averaging about 20 times the strength across the grain in the dry condition, and with its modulus of elasticity 15 to 20 times as high. Utilization of the full tensile strength of solid wood parallel to the grain is made difficult, however, because of the low shearing strength in this direction. Wood also is subject to unequal shrinkage and often warps as a result of moisture changes. The purpose of plywood is to meet these deficiencies by cross-banding, thus producing a homogeneous material having in any direction about half the tensile strength of wood along the grain, and no shrinkage.

Sheets of veneer cut from large logs are assembled and glued together with alternate plies at right angles to each other so that what is lost in one direction is gained in the other. All defects present in the original log are cut out of the veneer by large clippers before the veneer is jointed, sorted, and laid up to required size. Ordinarily, panels are three-ply, although as many as 25 plies have been used. The ultimate thickness depends upon the veneer thickness and number of the plies.

Besides the wood fibers, plywood is dependent for its strength on the adhesive material that binds the veneers together. Practically all species of wood in the United States may be glued successfully with one or more classes of adhesives: blood albumin, a by-product of the packing industry; casein, made from milk; vegetable proteins; and most recently synthetic resin. For the vegetable variety the soya bean is extensively used, producing a quality of glue similar to casein.

Albumin glue becomes 95 per cent insoluble under heat and pressure. Casein glue at room temperature becomes more or less water resistant through chemical changes that take place during the long drying period. Phenol formaldehyde resinoid binder is similar to blood albumin in that it is thermo-setting, requiring both heat and pressure. The resulting resinoid panel has 7 to 10 per cent moisture content and is highly water resistant. High shear strength and increased flexibility are claimed. U. S. Government specification shearing tests on blood albumin glue indicate that the average strength of a glued joint in shear is 370 lb. per sq. in. dry, and 240 lb. per sq. in. after being soaked for 48 hours.

Forty-seven known species of veneers are in use in wood-consuming industries of the United States. However, the industrial importance of many of these is slight, and they are used for purposes of no interest to the product engineer, being chiefly lumber substitutes. Excluding these, 95 per cent of the annual consumption of 35,000,000 equivalent bd. ft. of plywood is made up of 14 species. Of this Douglas fir constitutes 26 per cent of the total, with poplar and gum next in volume.

Tests made by the Forest Products Laboratory have established that the maximum strength of solid wood varies directly as its density. However, the lighter plywoods generally are stronger per unit of weight than the high-density species. Weight for weight, almost any plywood is stronger than steel both in tension and transverse tests.

Metal-faced plywood is a natural development, being made by glueing one or two metal faces to the plywood panel, thus producing a panel light in weight yet extremely rigid and strong. A $\frac{5}{16}$-in. panel of this material, for example, weighs the same as 16-gage sheet metal ($2\frac{1}{2}$ lb. per sq.ft.), yet it has approximately two times the stiffness and six to eight times the impact resistance of the 16-gage sheet metal. To assure a flat surface free from wrinkles, a stretcher-leveled sheet is used, usually 24 or 27 U. S. Standard gage. The material generally selected is a copper-bearing steel, electrolytically coated with zinc and hence rust resistant. Other metals, such as copper, stainless steel, Monel metal, and Everdur, have been used in conjunction with plywood.

With metal facing, plywood takes

on the additional properties of fire-resistance, a smooth surface for the application of finishes, and a means of completely sealing the wood interior against the entrance of moisture.

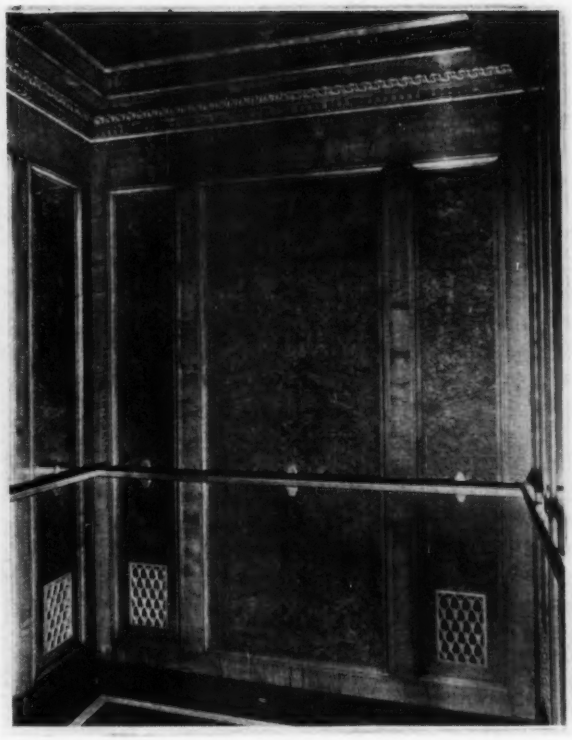
The kind of wood used for plywood varies with its use; two or more species can be used in the same panel. In the latter case, faces and backs are hard or high-density woods, such as birch, gum, mahogany, spruce, maple and hard poplar, while cores are soft or low-density wood such as balsam, basswood, fir, chestnut, gum and soft poplar. For use with metal facing, poplar, gum and Douglas fir panels are usually used.

Plywood and metal-covered plywood can both be permanently bent; the former may be molded into curved forms, provided a water resistant glue, such as blood albumin, is the binder. Plywood can be bent in one plane by soaking it thoroughly in hot water and then wrapping it around forms to the desired shape. Where bending takes place in two planes or where a warped surface is produced, it is necessary to mold the material between dies in a hydraulic press. Metal-covered plywood may be bent in simple curvature by passing through bending rolls, as is done with sheet metal, or between dies under hydraulic pressure. Little has been done, however, in molding metal-covered plywood.

Plywood is a definite engineering material and may be calculated for a stress-carrying member the same as steel or any other homogeneous material. It should be borne in mind, however, that the modulus of elasticity of wood is approximately 1,600,000 lb. per sq. in. along the grain as against 80,000 lb. per sq. in. across the grain, these being the values for poplar or Sitka spruce. Hence, in calculating the stresses at extreme fibers, it is not possible to use the ordinary beam formula, $p = My/I$. Since the sections are not homogeneous, the section modulus cannot be computed directly. Instead, the formula is recast to read: $P = \frac{M\,Ey}{\Sigma\,(EI)}$, the $\Sigma\,(EI)$ being determined for the whole section by adding the EI of each part. In every case, the value of E must be that for the material in that section and corresponding to the direction of grain, and y is always the distance from the neutral axis to the extreme fiber of the section under consideration. In finding the extreme fiber stress, the value of E for the part in question is substituted in the numerator, the EI value for the entire section is used in the denominator. To estimate the bending moments required to break a panel, it is safe to assume that the strength of wood fibers parallel to the grain is 10,000 lb. per sq. in. The lesser strength of wood across the grain does not appear to affect the strength of a plywood panel.

With metal-covered plywood, the calculation is complicated to the extent that additional material, steel, with a different modulus of elasticity is introduced, but the calculation may be made by taking each factor into account separately, performing the calculations in a routine manner.

Plywood has become a standard fabricating material in many classes of products such as motor vehicles, trunks and valises, fixtures, musical instruments, refrigerators, kitchen cabinets, airplanes, electrical machinery and apparatus, toys and games, motion picture scenery, ships and boats, elevators, clocks, sporting goods, sewing machines—in fact, wherever light, split-proof, warp-

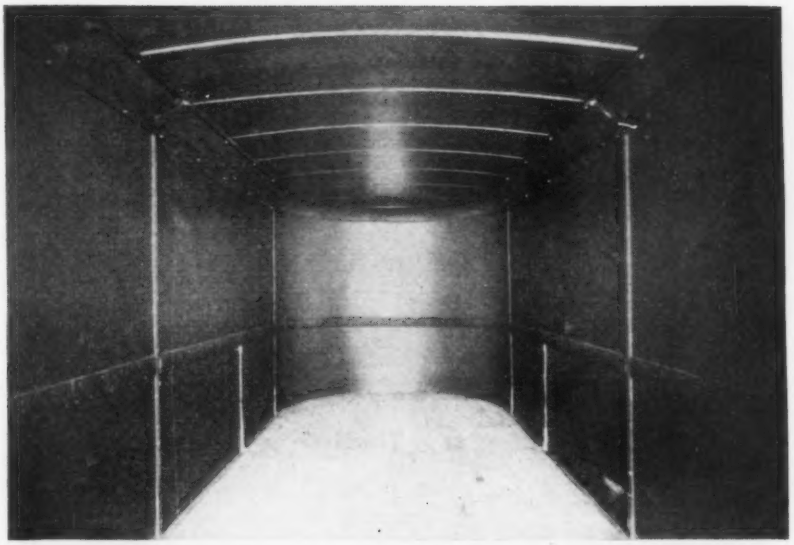


Plywood panels faced with the finest hardwoods line the Otis elevator cabs in the Waldorf-Astoria Hotel

Specific Gravity and Strength of Plywoods

(after Elmendorf)

Plywood Specie	Specific Gravity	Shearing Strength (Lb. per Sq.In.)		Column-Bending Strength (Lb. per Sq.In.)		Strength Under Tension (Lb. per Sq.In.)	
		Parallel to Grain	Perpendicular to Grain	Parallel to Grain	Perpendicular to Grain	Parallel to Grain	Perpendicular to Grain
Basswood.....	0.42	1260	1370	7120	1670	6880	4300
Pine (sugar)..	0.42	1310	1420	8050	1670	5430	3690
Chestunt.....	0.43	1250	1420	5160	1110	4430	2600
Cypress (bald)	0.45	1200	1370	8890	1850	6160	3980
Douglas Fir...	0.48	1080	1290	9340	1940	6188	3910
Mahogany....	0.48	....	....	8500	1940	6390	3780
Yellow Poplar	0.50	1530	1700	8860	1920	7390	4720
Gum, red.....	0.54	1500	1820	9970	2070	7850	4930
Walnut(black)	0.59	1560	1670	12660	2770	8250	5260
Oak (white)...	0.64	1340	1580	10490	2310	6730	4200
Birch (yellow)	0.67	2220	2350	16000	3200	13210	7700
Maple (hard)..	0.68	2260	2500	15600	3340	10190	6530



Photographs by courtesy of the Haskelite Manufacturing Corporation and the Harbor Plywood Corporation

Above—Full utilization of all the interior space without frequent supporting ribs is possible in this Fruehauf trailer van made of metal-covered plywood. Weight saving is an important gain over standard construction. Below—Plywood gives added strength and rigidity to taxicab and passenger car bodies. Factors of utility predominate over those of appearance

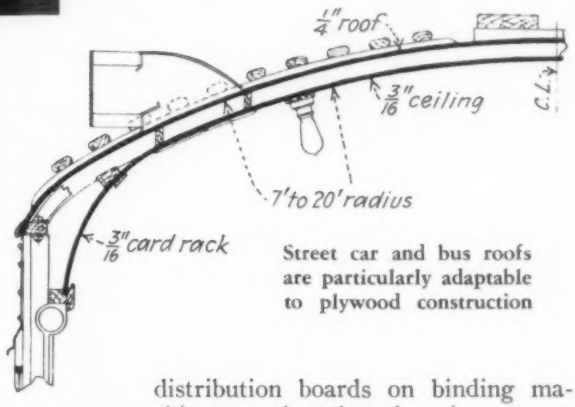
resistant material is needed. In addition to the conventional and standard uses of plywood, there are many specialized uses requiring curved, impregnated, composition-covered, and metal-covered plywoods. Impregnated or specially surfaced plywoods will resist against chemicals, steam or other unusual agents of deterioration.

Although most plywood is highly water-resistant, it is not wholly waterproof since wood always tends to absorb water and swell. Consequently, in exterior uses of plywood such as truck body panels, the edges should be embedded in grooves or rabbets filled with lead and oil, and the surfaces protected from the weather. Exterior sides are usually finished with standard lacquer or varnish. Aluminum coatings give the best protection. Roof panels for buses are covered with light canvas, glued down and painted. For ship use, panels are impregnated with hot paraffin, asphalt, or asphalt paint under pressure. Sometimes these panels are covered with cloth when in place, the cloth being held by marine glue and finished with shellac and spar varnish, or marine paint. Before the finish is applied, the moisture content of the panel must be low, not exceeding 8 per cent.

In aircraft construction plywood serves for covering fuselages, wings and tail surfaces. For this purpose, $\frac{1}{16}$-in. 3-ply mahogany is the common choice. The same material also is adaptable for small gusset plates in rib construction, and for the webs of wing beams. For cabin linings and floorings in aircraft, a balsa wood core, faced with aluminum or duralumin, makes a light panel that insulates against noise and heat.

Both standard and metal-faced plywood are found in buses, street-cars, ships and elevator cabs. In buses and street-cars, the chief applications are for roofing, head lining and card racks, seat backs, and interior trim and frieze panels.

By routing a 90-deg. V in the back side of single-faced metal plywood, it is possible to form square corners without a break. Unit refrigerating cabinets and moth-proof vaults are made in this way. Applications of standard plywood also include the bases of washing-machines.



Street car and bus roofs are particularly adaptable to plywood construction

distribution boards on binding machinery and outboard racing motorboat hulls.

Nails, tacks, or screws can be driven in the face of plywood without fear of damage by splitting. Metal-covered plywood is fastened with screws or cement-coated nails, and is covered with wood or metal molding after the panel is in place. It can be fastened to steel frames

by hot or cold riveting without danger of excessive charring of the wooden core. Rivet heads are driven against metal washers or cover strips not less than $\frac{1}{8}$ in. in thickness.

Thin sections of plywood lend themselves to molding in intricate art forms between embossing dies. With the true grain effect preserved, the result is strikingly similar to carved wood. Such a molded plywood, known as "Karvart," finds its chief use in radio cabinets, door panels and wall material and for elevator cabs.

Plywood is a relatively inexpensive material. Aside from the saving in direct material cost, it has a number of other advantages with respect to low-cost production. Being available in sheets of large size, plywood eliminates the ordinary glueing and joining of solid boards, and production proceeds rapidly without waiting for joints to set. Plywood comes in smooth surfaces, requiring little attention to secure a finish for polishing or painting. It can be had from the producing mills molded to form, cut to dimension, or specially treated ready for final assembly, saving not only in shipping weight, but in fabrication time at the assembly plant. The increasing importance of this material is shown by the fact that the volume of plywood produced has increased more than 70 per cent within the last six-year period for which figures are available.

[Editor's Note—*Grateful acknowledgment is made the Douglas Fir Plywood Manufacturer's Association, the Haskelite Manufacturing Corporation and Mr. George Gage of McCann-Erickson, Inc., for their thoroughgoing cooperation in the preparation of this article.*

Dowel Embossings for Assembling Stampings

(*Continued from page 414*)

of the parts have a hole and one part has the stamped dowel. It should also be noted that the parts vary in thickness and that the stamped dowel in this sort of combination should be located in *A*, the thickest part. This is done to enable the dowel to be made long enough to pass through the second part *B* and into *C*, the third part.

The dimensioning and variations for this type of design are similar to those of the previous examples. Care should be exercised in placing dowels so they are not too near the edge, or a bulging on the outside may result on account of the stock being forced out.

In another arrangement answering the same purpose, shown in Fig. 10, we have two thick parts held together and a third part, much thinner, adjacent to one of the other pieces. This problem involves a double set of stamped dowels where the punch impression has to be used as an alignment for the stamped dowel in the other heavy part.

The detail dimensions given in Fig. 11 illustrate a double stamped dowel design that locates three parts in relation to each other, with the heavy parts adjacent to one another.

Referring to *A*, Fig. 11:

The diameter of the punch impression *D* is equal to the diameter of the dowel *D*1 plus approximately $\frac{1}{2}$ *T*.

The diameter of the dowel *D*1 should not be made less than $\frac{1}{2}$ *T*.

Length *H* of the dowel in this type of design should be made less than $\frac{1}{2}$ *T*. In heavy stock this may be made as small as $\frac{1}{4}$ *T*.

The depth of the swaged impression *P* is equal to the amount of metal necessary to fill the dowel portion bounded by *D*1 and *H*. It should be noted that the rounded type of designed punch press impression is made use of in this part.

Referring to *B*, Fig. 11:

The diameter of the punched impression *D*2 must be made larger than *D*1 to assure dowel *D*1 entering *D*2. A good practical allowance suitable for most designs is $D2 = D1 + 0.0015$ in.

The diameter of the dowel *D*3 can be made the same as *D*1. This diameter should always be made at least 0.001 in. smaller than *D*2.

The length of the dowel *H*1 should in this type design be made to correspond with the amount of stock replaced by the punch impression bounded by *D*2 and *P*1.

The depth of the swaged impression *P*1 must be sufficient to allow dowel length *H* on first part to enter impression on second part. It should be made less than $\frac{1}{2}$ *T*1. This impression *D*2-*P*1 serves two purposes, it provides the metal for the dowel *D*3-*H*1, and it is also the locating hole for the dowel *D*1-*H*. In this case the metal in the impression is swaged from a regular cylindrical portion, and is one reason why it is desirable to have a larger diameter impression than the dowel diameter.

From *C*, Fig. 11, we have:

The dowel locating hole *D*4 is made of sufficient size to allow the dowel *D*3 to enter. In the design a good practical allowance is normally a minimum of 0.001 in. larger. If the relation of parts to one another is extremely important, the hole could be held closer by shaving to size as desired. Another factor to consider is not to make the hole *D*4 less than *T*2.

In this type of stamped dowel design, especially where there is more than one dowel involved, it is important that the dowels be accurately located.

Fig. 12 illustrates an alternative method of using two dowels and one hole.

For stock up to approximately $\frac{3}{16}$ in. in thickness these rules, from past experience, make good designs. For greater stock thickness it is sometimes necessary to increase the size of the depressed cavity in greater proportion to the size of dowel required and decrease the depth of the cavity.

IN THE NOVEMBER NUMBER

A complete summary with critical analysis and authoritative comments on the results of a nation-wide investigation conducted by *Product Engineering* covering current activity in product development, segregated into 12 fields, each treated separately, and setting forth:

Present status and character of product development activity.

Design factors being emphasized.

Trends in the adoption of new materials, unit parts, types of construction, and finishes.

The result of more than 2,000 questionnaires, personal visits by members of our editorial staff, and the cooperation of 20 associated McGraw-Hill publications, this survey gives a most reliable and detailed picture of what the other fellow is doing.

COMMUNICATION AND COMMENT

The Ideal Friction Clutch

[Editor's Note—*As stated by the author of this satire, who is the chief engineer of one of the leading clutch manufacturers but does not wish to have his name divulged, the subject matter "is purely a satire, the unfortunate part being that every remark is based on truth." Undoubtedly, many of our readers will feel that the same general situation pictured here, also pertains to their lines of products.*]

It being reluctantly conceded by engine manufacturers and machine builders that a friction clutch is, unfortunately, a necessary adjunct to their product, it becomes essential for the clutch manufacturers to meet the conditions they specify. A friction clutch is necessary in order that coacting parts may be coupled together or disconnected, and the clutch must accomplish this without causing chatter, vibration, loss of power, or being in any way detrimental to the efficiency of the machine to which it is applied. Based on many years experience, the specifications herewith have been drawn up, and are submitted as embodying most of the requirements considered necessary.

As the clutch couples coacting parts, it must be possible for it to pick up and carry the load at any desired speed, without noise or vibration, and at any degree of slip that the operator considers essential. Having brought the two parts up to identical revolutions, it should hold them positively without any greater variation than 0.0001 per cent in speed. It should act automatically as a governor, and should anticipate any trouble that might arise, disconnecting itself immediately when such occurs.

No parts of the clutch proper should be subject to wear, and therefore no adjustments of any kind should ever be necessary. All metallic frictional surfaces should be of Carboloy. Widia metal, or similar substance, and all parts should be proportioned so that the original dimensions and tolerances remain constant. Though this has not yet been achieved, research on wear is now being conducted at the expense of clutch manufacturers to develop such a desirable feature. Pending the introduction of a wearless disk, arrangements should be made for a magazine attachment that will install new disks as required, merely by pressing a button or similar device.

The clutch should be inherently elastic, and should be capable of expansion or contraction to suit the ideas emanating from the often nebulous mind of the designer. A clutch to carry 20 hp. per 100 r.p.m. should be capable of being installed in a space not to exceed 2x4 in., or as an alternate, the space occupied should not be more than ¼x6 in., merely by extending the plates.

Flexibility of installation is an important factor. The clutch should be able to carry any load, irrespective of rated capacity, and must carry this load in any position. It should be possible for the clutch to operate easily and at its maximum efficiency with an angular misalignment of 30 deg., and a lateral displacement of 1 in. in any direction. It is presumed that, although these limits are extremely close, it should be possible to hold them.

As inertia and momentum sometimes interfere with the designer's ideas, it is desirable to abolish these two factors, and thereby not require any means for braking.

The clutch should be designed to maintain a constant coefficient of friction which should not vary more than + or —0.0000001 per cent between the limits of —100 deg. F. to + 1000 deg. F. Dissipation of heat accidentally generated should be controlled by thermostatic means, and at the most suitable temperature to insure maximum efficiency. Where no rise of temperature is permissible, means should be provided for cooling the plates automatically.

In order to avoid any manual labor on the part of the operator, the clutch should be operated by thought waves emanating from the mind of the operator, though in some instances this might be difficult owing to the impossiblity of producing such waves in a vacuum.

More About Bolt and Nut Standards

W. S. Brown

In his remarks, in the July number of *Product Engineering,* George S. Case, president of the American Institute of Bolt, Nut and Rivet Manufacturers, appears to have misunderstood some of my remarks that appeared in the April number, relative to applying accepted standards. He suggests that 99 per cent of the nuts and bolts produced in this country have the American Standard Thread. This may easily be so because for all practical purposes this is the same as the United States Standards thread. What the writer had in mind in his previous discussion was the standard sizes of hexagon and thickness of the nuts and bolt heads. For some time these were difficult to obtain in certain sizes which differed from the S.A.E. standard of that time or were above the range of automotive sizes. We are now able readily to obtain sizes to and including 1 in., but beyond this size we still have the difficulty of rather poor deliveries.

Apropos American Standard nuts, only very recently we had to accept square bottomed slots in order to get good deliveries from a prominent manufacturer. Both the S.A.E. and American Standard illustrations of nuts with slots show the latter round bottomed. It is not that the

square bottomed slots were not satisfactory for the job, but the lack of uniformity and standardization is irksome.

The same manufacturer, and at least one other, carries stocks of slotted nuts of U. S. standard hexagon the slots of which seem to conform to some manufacturers' standard. Now many of these slots are quite satisfactory but there is a range, including $\frac{5}{8}$ in. to $1\frac{3}{8}$ in. nuts, in which the slots have been cut too wide.

Inquiry has elicited the information that as yet there is no standard for slots in the U. S. standard hexagon nuts which is recognized by either the Bureau of Standards or American Standards Association. Would not this be an opportune time for the Nut and Bolt Manufacturers' Association to sponsor vigorously a campaign to standardize a slot for both the American Standard Regular Nuts and the U. S. standard nuts?

The writer understands that the latter are now tentatively adopted by the American Standards Association as the American Standard Heavy Series. As the diameter of the cotter pin, and therefore the width of slots, is dependent on the bolt diameter, why not adopt the same dimensions for these as now used by American Standard Castle nuts and S.A.E. slotted nuts? Having gone so far, would it not be logical to take over also the dimensions for the depths of slots from the latter, although admittedly they are, in some cases, rather shallow? A not inconsiderable advantage to this would be that firms who may use both U. S. standard hexagon nuts and American Standard Regular Nuts slotted, either all the time or during the transition period from one standard to the other, would be able to use the same stocks of pre-drilled bolts and studs for both types of nuts, due allowances being made for lengths.

Discussion of Lettering Standards

A. W. Swain

As to his remarks in the August number, I agree with Mr. Shanks on the following statements: Vertical or slant lettering does not appear the same when either is produced by two or more men. Enforcing a standard style of lettering results in time consuming methods if the lettering is done either free-hand or with guide lines.

I disagree with Mr. Shanks on the following statements: I do not believe that it limits the individuality of the draftsman, cramps his self-expression or curtails the benefits of his experience. I fail to see where the type of lettering will curtail the benefits of a draftsman's experience any more than a certain type of penmanship with which a good toolmaker fills out his job time-card would curtail the benefits of his experience. Some draftsmen seem to have the temperament of a grand opera prima-donna when anybody tries to get any real action from them.

In conclusion, I would like to say that I firmly believe in a standard type of lettering, provided it is produced in a manner which is more economical than otherwise, and produces blueprints which are more readily understood in the shop, which in the final analysis is a real requirement for drawings.

We are using a standard type of lettering using capitals only and produced uniformly by a standard type of lettering guide which is on the market at the present time. We adopted this standard for three reasons: First, the lettering is more legible in the shop; secondly, the draftsman puts more time on the actual design, layout and detail, instead of spending a lot of unnecessary time in producing legible lettering; third, a tracer can take the drawing and produce a tracing with clear, legible lettering at a cheaper cost than by other methods.

Motion Study in Machine Design

Robert Pash
Mechanical Engineer

I have read with much interest Mr. Mogensen's article, "Motion Study in the Design of Production Equipment," which appeared in the August number of *Product Engineering,* and also Mr. Bullard's comments in the September number.

The subject under discussion is of particular interest to me as a machine designer because of my recent experience with a type of special production equipment which was found to be idle almost 90 per cent of the operation cycle. While fully aware of this fact at the time the machine was being designed, the intricacy of the machine prevented me from seeing any remedy. A motion study of the process was then taken and projected one frame at a time. This study, made with a background of motion study principles, readily gave the design requirements for a new machine. Although to meet these requirements necessitated considerable design work, the new machine proved to be only slightly more complicated than the former one, and idle time was reduced to a small fraction of the former figure.

A point I should like to emphasize is that motion economy, with its resultant increased hourly output, affords not only labor saving but also a reduction in machine cost. Cutting down the time used by the operator in preparing the machine represents a saving in both the operator's and the machine's time. One can visualize a case wherein this saving might make unnecessary the purchase of additional equipment.

Mr. Bullard's contention that separate attachments increase the cost of a machine and are likely to detract from its appearance, seems to be well founded. However, at the time of laying the ground work for a new machine the designer is not hampered by existing conditions such as confront the user. The designer can completely rearrange the controls and frequently incorporate labor saving devices without adding materially to the complexity and cost of the machine.

To decide as to whether a particular motion saving feature is worth having on a standard machine calls for keen engineering judgment and observation. Still, in general, as long as a machine is going to be operated by a human being, any inherent motion economy feature will be of value to the purchaser, regardless of the nature of his product.

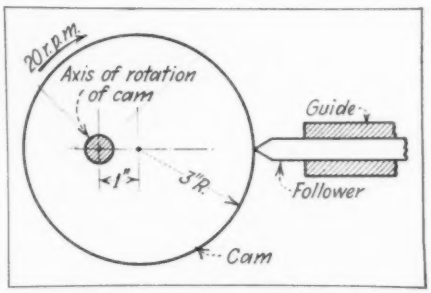
The machine builder, it appears to me, is in a better position to develop reliable mechanisms than is the user. The user can afford only a small development expense as he is merely trying to meet one particular production difficulty. The machine builder, on the other hand, can afford to ex-

periment, gather data and construct trial machines. Once he has produced an advanced design he can market it throughout the industry. Most mechanical features, including motion saving devices, are expensive to develop but once perfected can be manufactured at relatively low cost.

A Problem in Cams

W. W. JOHNSON
Trundle Engineering Company

In the course of the design of a special machine, I found it necessary to determine the maximum rate of acceleration of the cam follower as indicated in the accompanying illustration. After a considerable amount of labor, I succeeded in arriving at an analytical solution, but it involved some rather intricate mathematics. I have often wondered whether this problem cannot be solved in a much simpler manner, because on the face



What is the maximum rate of acceleration of the follower?

of it it appears to be extremely simple. I will be interested in seeing some of the methods for solving this problem as might be proposed by readers of *Product Engineering*.

Clarity in Dimensioning

ALFRED C. LAYZELL

I have no patience with the man who makes a "trick drawing" which is accurate in every way, but is misleading to the average machinist. A little more clarity and simplicity of the drawing might save a vast outlay of both money and material. Simplicity may be the hardest thing in the world, but that is what is required in the drawings for the machine shop. The following are a few suggestions towards that object:

1. Every item should have its dimensions given so that there will be no calculation for the machinist to do. Often a series of dimensions is given, and the final one omitted, which is to be calculated from the over-all dimension. This may be the first dimension required for the machinist, and he may start off the job with an error.
2. There should be no necessity to refer to conversion, gear, drill and tapping tables, a misreading of which might cause an error.
3. The chief draftsman should insist on uniformity in drawings. Dimensions should always be given in the same manner.
4. Slots with radius ends should always be dimensioned from one like point to another like point, and not from the center of one slot to the radius center of the other slot.
5. If one hole, counterbore, or slot in a series is different from the others, a note should be made calling attention to it.
6. Length of studs should be expressed as the length of full diameter required, not including the radius of end.
7. All angles should be given from one direction only.
8. When dimensions are given as ½" —0.005, the operator often thinks that he has to make this dimension actually —0.005, and as a consequence frequently runs under the low limit. If the dimension were shown as 0.500 +0.005 the operator would plainly see what is required.
9. With dimensions given in feet and inches, it is well not to use the sign (′) for feet, as sometimes the sign is not seen clearly. Such a dimension as 1′1″ may be read as 11 in., whereas if it were written as 1 ft. 1″, there could not be a mistake.
10. If the metric system is used for any dimension, it should be used throughout the drawing. If that is impossible, any other standards should be so marked that the machinist will be certain to notice it.
11. Root diameters should be shown as well as depth of teeth. Similarly for both internal and external recesses, the depth that the machinist should feed in should be given as well as the diameter at the bottom of the slot.
12. The material to be used should be shown with great prominence. This will obviate the frequent scrapping of valuable work because the wrong material has been used.

Data Wanted on Worm Gear Drives

One of the many services of *Product Engineering* to its readers is to assist in every way in the collection of data and information that might be useful in solving current engineering problems. We therefore take pleasure in publishing in full the following communication which was addressed to E. K. Ambler, the author of the article referred to in the letter:

Dear Sir:

Your article in the February issue of *Product Engineering* on safe loads for worm gearing has been called to my attention.

The Worm Gear Committee of the A.G.M.A. is attempting, by means of collecting existing data and making test runs, to determine a formula for the load carrying capacity of worm gearing. This program has been in force for some years, but due to existing business conditions, very little has been accomplished in the past year.

The Committee is compiling data on successful drives such as the one you describe, and in order to analyze it correctly, it is necessary to know more details of the drive—the pitch and outside diameters of the worm, tooth proportions, pressure angle, number of teeth in the worm gear, materials, mounting (worm above or underneath gear), lubricant, and lubricating conditions (hand lubricated, force feed, or submerged in oil). If you have drawings of the worm and worm gear, they will be of some assistance.

If you can send me the data requested, it will be of great help to the Committee and sincerely appreciated.

If others knowing of successful drives would only call them to our attention, I am sure that it would only be a short while before the A.G.M.A. would be able to offer a very complete analysis on the subject of allowable loads for worm gearing.

Yours very truly,

(signed) T. R. RIDEOUT, *Chairman, A.G.M.A. Worm Gear Committee, Westinghouse Electric & Mfg. Co., 200 McCandless Avenue, Pittsburgh, Pa.*

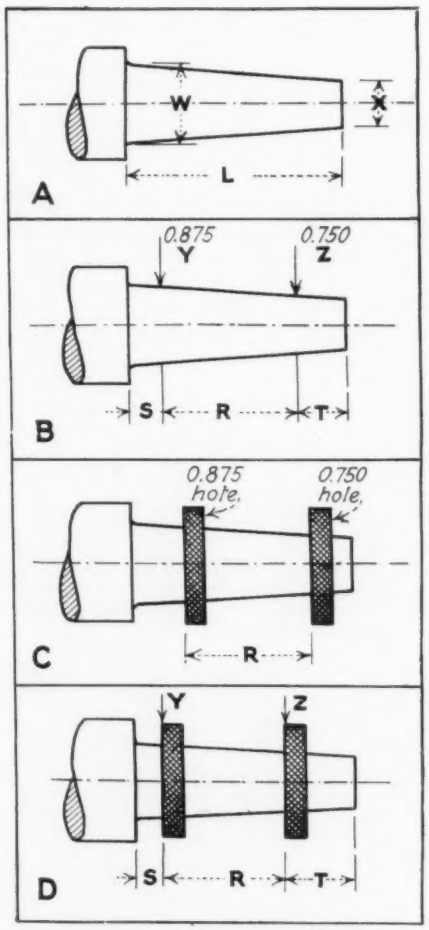
Particular attention is called to the last paragraph of the above letter, and we urge all of our readers who may be in possession of information that might be of assistance to the A.G.M.A. Worm Gear Committee to forward it to Mr. Rideout or to the editors of this publication.

Dimensioning Tapers

RALPH M. GAMBLE

Years ago, when the country was just going into a huddle after the World War and things were a bit dull, the writer was called upon to make permanent records of certain methods and gages that had proved to be accurate and highly successful. The methods were in the toolmaker's head and the gages were in his tool box.

These gages were for rifle barrel cartridge seat reamers and were tapered in two places, but for the sake of simplicity we will consider only one taper as in the accompanying illustration. At *A* is shown a method of dimensioning often employed by many draftsmen. While the figures may be right, it is difficult to measure them accurately for the reason that one cannot get a good micrometer reading on either end of the taper.



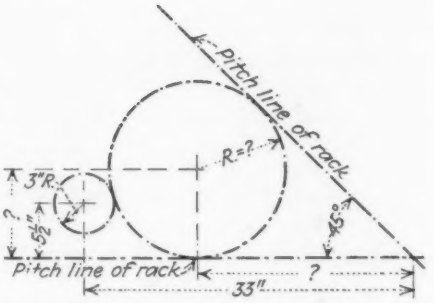
Illustrating a common method and the correct method for dimensioning tapers to facilitate manufacturing

At *B* in the illustration is shown what is unquestionably the right method to use if accurate results are required, as it gives the toolmaker plenty of room to get the proper taper angle, before finishing to size. All he need do is to make a set of washers reamed to the sizes called for at *Y* and *Z*. He may harden and grind them if the occasion warrants. When grinding the taper on the reamers, he can check the dimensions *R*, indicated at *C*, and thereby insure accuracy long before arriving at the final sizes as indicated at *D*. It is obvious that the dimension *R* will not vary while grinding to obtain dimensions *S* and *T*. This looks simple and is easy to understand when presented correctly, and therein lies the value of good drawings.

A Problem in Locating Centers

J. A. COVINGTON

In the design of a special machine, an unusual problem of locating a gear and calculating its diameter was encountered. Actually there were two gears of equal pitch diameter, one on each end of a cross shaft, and each gear driving a rack. What complicated the problem was that the pitch lines of the rack intersected at an angle of 45 deg. In addition, it was required that one of the gears also mesh with a 6-in. diameter pinion.



What is the diameter of the gear and the location of its center

Fundamentally, the problem resolved itself into determining the location of the center of a circle that would be tangent to the pitch lines of the racks and also tangent to the pitch circle of the pinion. Though the problem appears simple, it caused us many "sleepless nights" until one of the men discovered a solution that was as simple as the problem looks to be. Before submitting it, I would like to see what solutions readers of *Product Engineering* might offer.

DEFLECTIONS

GASOLINE FOR KEROSENE

Designers of internal combustion engines will be interested in the results of a survey made by engineers of the Ethyl Corp. According to their figures, 82 per cent of tractors designed to use kerosene are now being operated on gasoline. Tests have shown that a kerosene-burning tractor which will pull two 16-in. plows in ground-breaking operations will, when suitably altered and operated on anti-knock gasoline, pull three 16-in. plows at a speed 7 per cent higher.

BEARINGS WITHIN BEARINGS

"Wheels within wheels" is an old and trite saying; bearings within bearings is a proposition that has become an actuality. The idea is one of dividing the speed of a high-speed spindle between two bearings. One ball bearing is nested within another. The outer race of the inner bearing revolves with the inner race of the outer bearing, the speed of the spindle being divided between the two bearings. Such an installation has been made with success on a high-speed shaper for woodworking. The spindle has a speed of 18,000 r.p.m., each ball bearing turning at about 9,000 r.p.m.

MOULDING THE BEAD

The appearance of welding beads often detracts from the looks and lowers the salability of many an otherwise fine product. But designers can now specify automatically welded beads to be form molded to uniform size and smooth surface. It is reported that one tank manufacturer has been obtaining such results by attaching to the automatic head of the welding machine a shoe with a half round depression, thereby shaping the bead while it is still plastic.

GALVANIZED WELDED UNITS

The engineers of a large gas refrigerator manufacturer are now specifying galvanizing instead of painting for refrigeration units. After these units have been welded, they are galvanized at a temperature of 850. deg. F.

Malcolm Muir

A. W. Robertson

A.B.P. Undertakes Rehabilitation Campaign

At a meeting of the eastern members of Associated Business Papers, Inc., at the McGraw-Hill Building, New York, N. Y., Sept. 23, the personnel and plan of the A.B.P. Committee on Industrial Rehabilitation and Economic Recovery were announced. This committee, headed by Malcolm Muir, president, McGraw-Hill Publishing Company, publisher of *Product Engineering,* will work intensively in cooperation with the National Committee on Industrial Rehabilitation, headed by A. W. Robertson, chairman of the board, Westinghouse Electric & Mfg. Co. The National Committee, it will be remembered was appointed by President Hoover "to stimulate the expansion of capital expenditures by industry through the replacement of obsolete and worn-out equipment and otherwise."

The Robertson Committee with its sub-committees in every Federal Reserve district, in every line of business and in every manufacturing community in the United States, will concentrate on an effort to get manufacturers to modernize their plants now, at a time when their machinery is in a run-down condition, when financial and credit conditions are improved, and when purchase and installation costs for machinery are low. "The committee believes," said Chairman Robertson, "that if industry will proceed now for reasons of sound self-interest to put its house in order and to remedy through modernization the run-down condition of its productive equipment, employment will be created for hundreds of thousands of workers in the 'capital goods' industries. In addition, it will bring back on payroll hundreds of thousands of other workers [including engineers] in industries which furnish the parts, raw materials and services bought by the machinery and equipment manufacturers. This will release millions of dollars in wages spent for individual and family purchases affecting business in every community . . . "

In the industries devoted to the making of machinery and equipment alone, approximately 1,620,000 workers have been deprived of employment since 1929. Their loss in income totals more than three billions of dollars annually. It has been estimated in this connection that loss of employment for one worker in these "capital goods" industries results in loss of employment for three workers engaged in the marketing of consumer products—food, clothing, radio, automobiles—or in furnishing materials to the equipment and machinery builder.

The acuteness of this situation is further intensified by the fact that, along with the prostrating sag in employment, expenditures for equipment, machinery and plant facilities have dropped from the average normal annual outlay of $5,000,000,000 to a yearly total of $1,260,000,000. This falling off constitutes one of the major factors contributing to continuance of the paralysis which besets American industry.

The progress of industrial maintenance has been interrupted to such a degree that today more than fifty per cent of the machinery, equipment and plant facilities in American factories is obsolete. The fact that in these last three years there has been more rapid improvement in equipment design than in any period in our industrial history makes the condition that much more urgent. Replacement has not kept pace with this engineering advance.

The primary objective of the Committee on Industrial Rehabilitation at the present moment is to set in motion a succession of orders for improved equipment and machinery which will extend, in their effect on employment and spending, from the factory back to the farms, the forests and the mines. Every order starts a wave of work required in the production, processing, transportation, installation and sales of materials and services that is felt throughout all branches of business. The Committee on Industrial Rehabilitation in its national program is stressing the advantage to the manufacturer in urging him to proceed at once with the modernization of his plant.

The committee is confident that if industry will proceed at once for reasons of sound self-interest to put its house in order, employment will be created for hundreds of thousands of workers in the "capital goods" industries.

Engineers' Council for Professional Development

As the result of meetings in the past several months of representatives of seven professional engineering organizations, including the National Societies of Mechanical, Civil, Electrical and Chemical Engineers, and the Society for the Promotion of Engineering Education, the Engineers' Council for Professional Development has been organized. The ultimate objects of the organization have been listed as follows: to promote higher professional standards of education and practice in engineering, to bring about greater solidarity of the profession and to make the engineer more effective in dealing with technical, social and economic progress.

An immediate objective is to set up procedure and qualifications by which the progress of the engineer toward professional standing can be recognized by the public, the profession and the man himself. The plan comprehends a more or less automatic certification of the engineer in terms of at least minimum professional standards.

Electroplating Research

For several years the New Jersey Zinc Co. has been actively engaged in electroplating research. Many hundreds of samples have been plated with various types of metal coatings —nickel, nickel-chrome, chromium direct, copper, brass, bronze, cadmium, silver, gold, platinum—to determine the proper plating method and to prepare samples for various aging and service tests.

As the work in cooperation with several industries has expanded, the company has continually found it necessary to add new equipment, but until the present time this has always been of the laboratory type. Recently, however, this equipment has been augmented by the addition of tanks and a generator unit of a semi-commercial size. These new facilities should enable the company to develop and demonstrate on a practical scale various procedures for electroplating rolled zinc and zinc die-castings.

Wolf Trophy To Be Awarded

A second Irwin D. Wolf trophy will be awarded by the American Management Association and presented at the "Third Packaging, Packing and Shipping Conference, Clinic and Exposition" to be held in the Hotel Pennsylvania, New York City, March 7-10, 1933. Some modifications will be made in the rules which governed the award last year, the trophy having been designated for the best package developed and put on the market between January 1, 1932 and February 15, 1933.

Meetings

A.S.M.E.

The National Machine Shop Practice Meeting of the American Society of Mechanical Engineers will be held at the Hotel Statler, Buffalo, N. Y., Oct. 3, during the National Metals Congress & Exposition. Papers of interest to engineers and designers were listed in *Product Engineering* for September.

A.S.S.T.

The annual meeting of the American Society for Steel Treating will be held at Hotel Statler, Buffalo, N. Y., Oct. 3-7, in conjunction with the National Metal Congress and Exposition sponsored by the society. Engineers and designers will be interested in the following papers:

"Autographic Stress Strain Curves of Deep-Drawing Sheets"—Reid L. Kenyon and Robert S. Burns, American Rolling Mill Co.

"Design of Heat-Resisting Alloy Members and Their Performance in the Field"—Ralf S. Cochran, Surface Combustion Corp.

"Alloys of Iron, Manganese and Carbon"—Francis M. Walters, Cyril Wells, John F. Eckel and V. N. Krivobok, Carnegie Institute of Technology.

"Internal Friction in Iron and Iron Alloys"—Robert S. Canfield, U. S. Naval Research Laboratory.

"Corrosion-Resistance of Iron Chromium Alloys in Various Media—the Salt Spray Test"—Wm. B. Arness and J. N. Ostrofsky, Alloy Research Corporation.

"Allotropy in Iron & Steel"—R. H. Harrington, General Electric Co., and Wm. P. Wood, University of Michigan.

A.W.S.

The American Welding Society has released a definitive program of its 12th Fall Meeting to be held concurrently with the National Metal Congress and Exposition, October 3-7, 174th Regiment Armory, Buffalo, N. Y. The morning technical sessions and committee meetings will be held at the Statler Hotel, afternoon sessions will be held at the Armory. Papers of special interest to engineers and designers will be:

"Welded Rolled Steel Design for the Iron & Steel Industry"—Christian A. Wills, The William B. Pollock Co.

"On the X-Ray Determination of Stresses in Welds"—John T. Norton, Massachusetts Institute of Technology.

"Corrosion Fatigue Study of Welded Low-Carbon Steel"—W. E. Harvey and F. Jerome Whitney, Jr., Lehigh University.

"Distribution of Stresses in Welded Double Butt-Strap Joints"—S. C. Hollister and A. S. Gelman, Purdue University.

"The Application of Welding to the Manufacture of Heavy Mechanical Equipment"—A. E. Gibson, Wellman Engineering Co.

"Popularizing the Welded Product"—G. M. Gillen, G. M. Basford Co.

"The Chemistry of Low-Carbon Metal-Arc Weld Metals"—J. C. Hodge, Babcock & Wilcox Co.

S.A.E.

The Production Meeting of the Society of Automotive Engineers will be held Oct. 3, at Hotel Statler, Buffalo, N. Y., on the first day of the National Metals Congress & Exposition. At least two papers will be found of interest to engineers and designers:

"How the Design Engineer Views Manufacturing"—Alex Taub, Chevrolet Motor Co.

"Special Zinc Alloys in Die-Castings (with film)"—C. R. Mixon, New Jersey Zinc Co.

Immediately therefollowing, the Transportation Meeting of the Society of Automotive Engineers will be held Oct. 4-6, at the Royal York Hotel, Toronto, Canada. Engineers will find interest in the following papers:

"Ventilation of Motorcoach and Motor Truck Bodies"—W. J. Cumming.

"Refrigerated Motor Trucks"—H. M. Williams and J. W. Carl.

"Automotive Engines and Their Fuels"—Dr. Graham Edgar.

"Pneumatic Tires, Old and New (with film)"—Burgess Darrow.

"Six-Wheel Trucks"—A. M. Wolf.

Expositions

National Metal Congress & Exposition, 174th Regiment Armory, Buffalo, N. Y., Oct. 3-7.

Safety Congress & Exposition, National Safety Council, Washington, D. C., Oct. 3-7.

American Gas Association Convention & Exposition, Atlantic City, N. J., Oct. 10-12.

National Business Show, Grand Central Palace, New York, N. Y., Oct. 17-22.

Southern Textile Exposition, The Poinsett Hotel, Greenville, S. C., Oct. 17-22.

International Beverage Exposition, Public Auditorium, Cleveland, Ohio, Nov. 14-18.

National Exposition of Power & Mechanical Engineering, Grand Central Palace, New York, N. Y., Dec. 5-10.

Joint Trade Show, Motor & Equipment Manufacturers Association and National Standard Parts Association, Convention Hall, Detroit, Dec. 5-10.

Engineers and their Activities

W. C. Goodwin

William B. Mayo

WALTER A. BATES, B. BATES and CHARLES I. BATES have incorporated a company for the practice of mechanical and civil engineering under the name of Walter Bates Co., Gary, Ind.

CLYDE COATES, formerly designing engineer with the Taylor-Winfield Corp., Warren, Ohio, has joined the engineering staff of the Federal Machine & Welder Co., Warren, Ohio.

ROY E. COLE, formerly a partner in the Vail-Cole Engineering Co., has assumed the duties of chief engineer of the Rockne Motors Corp., Detroit, Mich.

W. C. GOODWIN, of the Supply Engineering Department, Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa., has been placed in charge of the engineering and development in the newly created department for the engineering, development and sales of air-conditioning apparatus.

A. B. HARGIS has been appointed head of the school of engineering of the University of Mississippi, University, Miss. For the past eleven years he has practiced as a consulting engineer in Jellico, Tenn.

ALBERT S. HEINRICH, of West Orange, N. J., recently severed his connection as engineer with the Bendix Research Corp., East Orange, N. J.

WEBSTER N. JONES has been elected director of engineering, Carnegie Institute of Technology, Pittsburgh, Pa. After fourteen years of industrial work, Dr. Jones thus returns to educational work in which he spent ten years of his career, teaching at Purdue University, Harvard University, University of Maine, University of Missouri, and University of Montana.

WILLIAM B. MAYO has retired as chief engineer of the Ford Motor Co., Detroit, Mich., and engineer in charge of the aviation activities of the Ford company as forecast in *Product Engineering* for September.

JAMES H. MCKEE, chief engineer, Pittsburgh Screw & Bolt Corp., Pittsburgh, Pa., presented a paper, "Hollow Steel Propellers," at the recent S.A.E. Aeronautic Meeting, Cleveland, Ohio.

GEORGE A. PAGE, JR., chief engineer, Curtiss-Wright Airplane Co., St. Louis, Mo., was co-author of a paper, "The Economic Aspects of Transport Airplane Design," presented at the Aeronautic Meeting of the Society of Automotive Engineers, Cleveland, Ohio.

R. F. PEO, vice-president and chief engineer, Houde Engineering Corp., Buffalo, N. Y., will preside at the first technical session on the program of the S.A.E. Production Meeting in Buffalo, N. Y., Monday, Oct. 3.

MALCOLM W. REED has taken up the duties of chief engineer of the American Steel & Wire Co., with headquarters in Cleveland. He succeeds B. H. LAWRENCE, who has been transferred to New York where he will serve as assistant to John Hulst, a vice-president of the United States Steel Corp.

EDWIN L. SMITH, formerly full-size-body draftsman with the Pierce-Arrow Motor Car Co., Buffalo, N. Y., has taken a similar position with the Studebaker Corp., South Bend, Ind.

S. M. STOLER, former chief engineer of the Porcelain Enamel & Mfg. Co., Baltimore, Md., has severed his connection with that company and has set up offices at 3103 Echodale Ave., Baltimore, Md., as a consulting engineer specializing in engineering work for the porcelain enameling industry.

F. L. TARLETON, formerly with Westinghouse Electric & Mfg. Co., Springfield, Mass., has been engaged as research engineer by Gibson Electric Refrigerator Corp., Greenville, Mich.

LOUIS E. UNDERWOOD, formerly managing engineer of the stationary motor engineering department of the General Electric Co., Lynn, Mass., has been appointed manager of the Pittsfield, Mass., works to succeed E. A. WAGNER, retired. Mr. Underwood was chief engineer of the department from 1915 until 1927, when he was placed in charge of both engineering and manufacturing.

GEORGE W. WALKER, 381 W. Maplehurst, Ferndale, Mich., has resigned as art director and design stylist for the Devereaux Corp., Detroit, to take up the duties of manufacturers' representative and industrial designer in Detroit, Mich. Prior to this he was designer of automobile bodies for General Motors and Graham Paige.

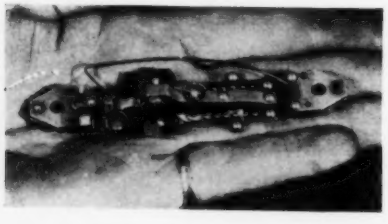
A. M. WOLF, consulting engineer, 18 Centre St., New York, N. Y., will present a paper on the design of six-wheel automobile trucks at the S.A.E. Transportation Meeting, Oct. 4-6, Toronto, Canada.

NEW MATERIALS AND PARTS

Baldwin-Southwark Scratch Extensometer

Weighing less than 1 oz., readily fixed to members under load—by rubber bands, by vacuum cups or by numerous other easy and convenient methods—"The Scratch Extensometer" is announced by Baldwin-Southwark Corp., Southwark Division, Philadelphia, Pa. A genuine white diamond, especially cut, is mounted so as always to be under a small, but adjustable load. It bears on a recording target of heat-treated steel. The strain in the member under test causes the diamond to scratch a record on the target. Simultaneously the target is under a resultant force, manually variable, to move at right angles both to the axis of the instrument and to the axis of the diamond. Hence, one coordinate of the record indicates strain while the other has no meaning but is availed of in order that a record may be had over a period of time more or less extended as fixed by the variability of the resultant transverse force and the amount and rate of strain variation.

When there is no change of strain there is no motion of the target; small changes in strain cause correspondingly small transverse movements of the target. Strains are evaluated from the record by filar microscopes which may be of the moving eyepiece or of the moving-table type. Or the record may be photographed microscopically and the print measured directly. With known magnification, the strain (and, therefore, stress) becomes determinable at once. From 100 to 500 diameters magnification is needed depending on the nature of the record made.



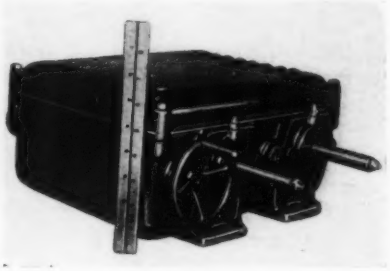
An important advantage of the Scratch Extensometer is that no multiplying system is needed on each instrument. This permits a great economy in manufacture and largely accounts for the low price of the instrument. One photomicroscope such as ordinarily used in metallography or one filar microscope suffices for the measurement of many targets. Targets may be filed away like correspondence. It should be further explained that with strain known and with the modulus of elasticity known, it is a very simple matter to evaluate stress.

Korfund Seismo-Damper

As shown in the accompanying illustration, the engineers of The Korfund Co., 48-15 32nd Pl., Long Island City, N. Y., have just introduced a device called the "Seismo-Damper" for the isolation of vibration and noise. This consists of a design whereby any isolating material (cork, rubber, etc.) inserted between an upper and lower pressed steel housing, is preloaded at the factory to exactly that point which compensates for the difference between the loading established by the operating machine and the peak efficiency loading required by the isolating material. It is claimed that the isolating material in the Seismo-Damper is always operating at its maximum efficiency. Ease of installation is an important factor; the upper housing is drilled and tapped for bolting directly to the base of the machine or the assembly for which it is designed.

Lewellen Variable-Speed Transmission

The Lewellen Mfg. Co., Columbus, Ind., announces its smallest size variable-speed, fully inclosed transmission to be known as size 106, series 100. Designed to meet conditions



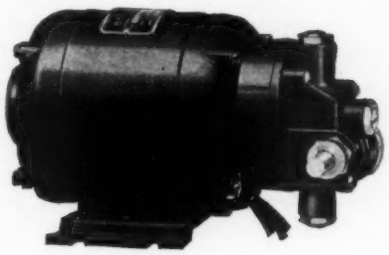
where dust, abrasives, oil, acids, etc. might injure the working parts of the open-type transmission, this unit is adapted for use in the drives of small production machines requiring accurate variable-speed control. The unit may be lubricated and adjusted without removing the cover or stopping operation. The series 100 fully inclosed type is built in all sizes from $\frac{1}{3}$ hp. to 15 hp. inclusive.

Bodine Motors and Speed Reducers

A new line of fractional horsepower motors with or without speed reducers is being marketed by the Bodine Electric Co., 2264 W. Ohio St., Chicago, Ill. This line includes the type NSY-12 self-starting, single-phase, synchronous motor in ratings of 1/75 hp. in the open-ventilated frame and 1/150 hp. in the totally inclosed frame at a motor speed of 1,800 r.p.m. Type NSI-12 induction motor is rated at 1/50 hp. in the open-ventilated frame and 1/70 hp. when totally inclosed at a motor speed of 1,725 r.p.m.; it is rated at 1/100 hp. in the open frame and 1/150 hp. when totally inclosed at a motor speed of 1,125 r.p.m. Worm gear speed reducers are available with the follow-

ing ratios: 8½-1; 17-1; 30-1; 35-1; 72¼-1; 144½-1; 289-1; 595-1; and 1120-1; other ratios are available if required.

These new motors are built in aluminum die-cast three-piece frames with separable gear heads, finished in baked black enamel. Nitralloy steel shafts running in sealed-type ball bearings are used for the rotor with the worm gear cut on the shaft. Laminated bakelite gears are used in the speed reducer. A typical motor weighs 4½ lb., has an over-all length of $6\frac{5}{32}$ in. and a diameter of $3\frac{21}{64}$ in; diameter of slow-speed driveshaft is $\frac{5}{16}$ in. All regular synchronous or induction types are furnished as capaci-



tor start and run when required. Shaded pole unidirectional or reversible units, four or eight poles, are also available in these frames.

"MK Steel"

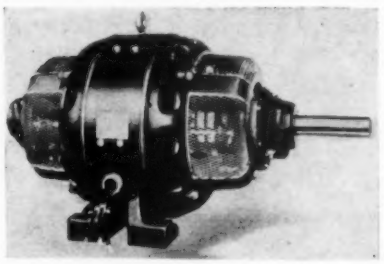
According to a recent announcement of the International Nickel Co., New York, N. Y., a new magnet steel has been developed in Japan to be known as "MK Steel." A typical composition is 25 per cent nickel, 10 per cent aluminum, balance iron, but the nickel may be varied from 10 to 40 per cent and the aluminum from 1 to 20 per cent. It is claimed that the coercive force is about nine times that of the usual tungsten magnet steel and 2½ times that of the best cobalt magnet steel. The residual magnetism can be varied so that its value is between that of tungsten and cobalt magnet steel.

MK Steels are not readily forged and must, therefore, be cast into the desired form. However, their desirable magnetic properties are developed by simple annealing and they are therefore free from the difficulties of quenching which is required with the tungsten steels. Magnet properties do not change appreciably with increase in temperature even to the extent of several hundred degrees C. MK Steels are also said to be stable magnetically when subjected to mechanical vibration or shock.

Ideal "Self-Syn" Motor

For general-purpose applications, the Ideal Electric & Mfg. Co., Mansfield, Ohio, announces a synchronous motor which is a compact, self-excited, self-synchronizing, self-contained motor unit requiring no external excitation. The exciter windings, direct-current commutator, etc., are built integral with the motor itself. Since no field switch or synchronizing relay is necessary, this motor is started with a simple hand-operated compensator in the same manner as an ordinary induction motor.

This "Self-Syn" motor will resynchronize itself automatically after it has been pulled out of step by a line-voltage dip or a momentary overload. Power factor correction is inherent because unity (100 per cent)



or 80 per cent leading power factor is standard. Motors are available in all sizes from 5 to 100 hp. at 900, 1,200 and 1,800 r.p.m.

General Electric Two-Pole D.C. Motors

Designed so that many variations, such as different degrees of inclosure for various applications, can be accomplished by using only a few different parts, a new line of d.c. motors is offered by the General Electric Co., Schenectady, N. Y. New and radically different pulley-end shields afford effective protection to the motor windings. Drop-forged feet welded to the motor frame reduce the possibility of breakage. Insulation has been improved to resist moisture and weak acids. It is claimed that properly proportioned stabilizing windings make possible unusually stable operation. In addition, the speeds of all constant-speed motors can be increased 25 per cent, and more in some ratings, by field control. We are informed that the new line is confined to a new type two-pole motor in ratings from ½ hp., 1,150 r.p.m. to 5 hp., 3,500 r.p.m. The standard line of G.E. d.c. motors is naturally still available.

Garlock Thiokol Packings

A number of packing items, such as diaphragms, disks and gaskets made of various Thiokol compounds are announced by the Garlock Packing Co., Palmyra, N. Y. Thiokol, which was given its first announcement in *Product Engineering* for April, 1932, is a new chemical product having most of the characteristics of rubber but unlike rubber not soluble in various hydrocarbons. The new Garlock Thiokol packings are advocated in place of rubber especially where they are to be employed in contact with gasoline, ether, kerosene, carbon tetrachloride or gas.

It is claimed that no sign of deterioration has been found after long periods of service for pipe flange gaskets and valve disks and packings in pipe lines conducting gasoline. Pump valves made of Thiokol have shown remarkable life in such service.

Merkle-Korff Power Reduction Unit

The new power reduction unit recently announced by Merkle-Korff Gear Co., 213 N. Morgan St., Chicago, Ill., is a matched assembly of gear reducer and induction (shaded-pole type) or synchronous motor.

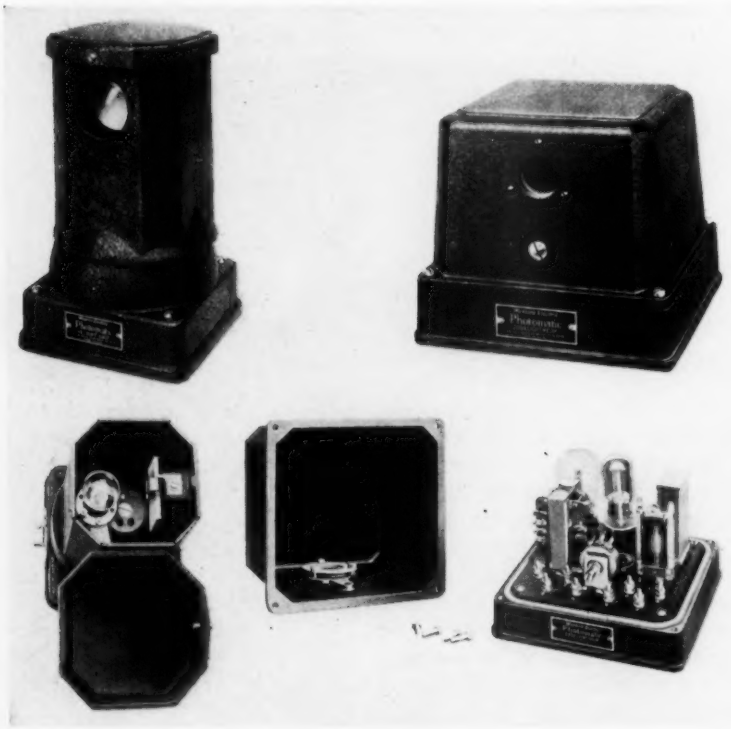
The gear housing is neat and compact; spur gears are sealed in grease, insuring positive lubrication and quiet operation. The unit is available in almost any speed from a motor speed of 3,000 r.p.m. down. It can be obtained for clockwise or counter clockwise rotation, and in both horizontal and right-angle drives. Right-angle drives can be had in any position within the 360 deg. circle. This "flexo-action" unit can be supplied with a maximum torque of 15 in.lb. in almost any desired speed. Over-all dimensions are as follows:

Horizontal Drive	*Right-Angle Drive*
Height, $3\frac{1}{8}$ in.	Height, $3\frac{1}{8}$ in.
Width, $3\frac{3}{8}$ in.	Width, $3\frac{3}{8}$ in.
Depth, $2\frac{5}{16}$ in.	Depth, $3\frac{7}{16}$ in.

Western Electric Photomatic Equipment

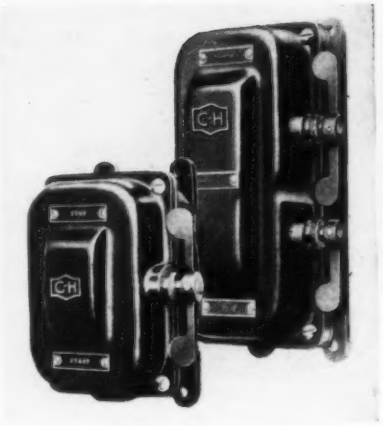
Designed by Bell Telephone Laboratories, and offered by the Western Electric Co., New York, N. Y., under the name of "Photomatic Equipment," a new photoelectric cell system consists of two small units, one the light source and the other a light-relay which contains the photoelectric cell. Both are designed to operate from commercial 50-60 cycle, 115-volt a.c. current. The connection of the a.c. current supply and of the circuit to be controlled by the photoelectric action comprise the only electrical connections required. The equipment may be arranged to open or to close an electrical circuit on either the interruption or completion of the light beam.

As shown in the illustration, the light unit is $5\frac{1}{8}$ in. square at the base and $8\frac{3}{8}$ in. high. It contains an ordinary automobile headlight bulb which is fed by a suitable transformer and furnishes the light for the beam as well as a condensing lens to produce an essentially parallel beam of light. The light relay unit is 7 in. square at the base and approximately $6\frac{3}{4}$ in. high. It contains a vacuum photoelectric cell of the caesium type, an amplifying vacuum tube, a relay, and associated apparatus. The cast aluminum housing has a window $1\frac{1}{16}$ in. in diameter to admit light to the cell. Additional aperture plates are provided with circular openings of $\frac{3}{4}$, $\frac{3}{8}$ and $\frac{3}{16}$ in. so that the amount of light admitted can be varied to meet different operating conditions.



Cutler-Hammer Lever-Operated Master Stations

For use under severe weather or atmospheric conditions, a new line of master stations has been announced by Cutler-Hammer, Inc., 296 N. 12th St., Milwaukee, Wis. Contrary to

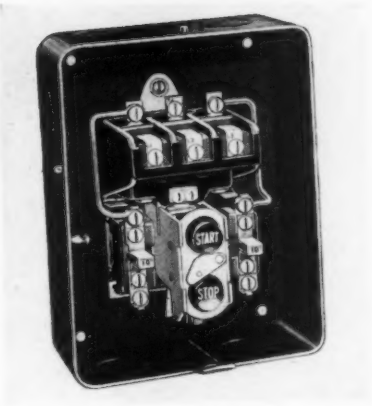


usual practice, these are operated from levers placed on the side of the case instead of from push-buttons on the front. It is claimed that by this construction an absolutely watertight case is secured. The shafts of the levers operate through stuffing boxes, and the cover joint is sealed by a rubber gasket. The stations use standard, heavy-duty, pushbutton elements, mounted on removable individual molded bases, fully inclosed. They are supplied in all normal ratings in one lever (on and off), two lever (start and stop), or three lever (hoist, lower, stop.)

Carpenter Stainless Steel with Selenium

When the free-machining 18-8 stainless steel known as Carpenter No. 8 was announced some months ago, it contained about 0.3 per cent of sulphur as a free-cutting agent. Now, however, the Carpenter Steel Co., Reading, Pa., announces that Carpenter stainless steel No. 8 will henceforth have a formula of approximately 18 per cent chromium, 9 per cent nickel, and 0.25 per cent selenium. The outstanding effect of selenium in stainless steel is to render it free machining; it has proven fully the equal of sulphur as a free-cutting agent and at the same time avoids some of the objections incident to the use of high sulphur.

So far as we can ascertain, this is the first time that selenium has been employed in ferrous metallurgy. It is said that when sulphur is added to a steel bath it combines with some metal to form a metal sulphide, such as manganese sulphide, zirconium sulphide, or molybdenum sulphide. These sulphides are entirely insoluble in the steel and occur in the cast ingot



as small non-metallic particles. They are quite ductile at forging and rolling temperatures and therefore during the hot working of the steel the sulphides are drawn out into long threads or stringers. Naturally these non-metallic stringers interrupt the continuity of the metal and tend to render the rolled bar much less ductile in a transverse direction.

Selenium, like sulphur, apparently combines with a metal to form a selenide, but these selenides are partially soluble in the steel. There is not therefore so much non-metallic matter to form stringers, and as a consequence the finished product is much less subject to tranverse fracture.

Tensile and impact tests made transverse to the direction of rolling on free-machining 18-8 stainless steel furnish a comparison between the effect of sulphur and selenium as free-machining agents. Holding all other conditions equal, the selenium steels have about one-third greater elongation and reduction of area and twice the Izod impact resistance. The ultimate strength is the same. Selenium steel is easier to roll and forge than the sulphur types, being less subject to splitting, cracking or opening up at the ends; it can also be upset with greater safety and less trouble from splitting. Selenium stainless steel is so freely machinable that it can be cut in automatic screw machines at 60 to 70 per cent of the speed of Bessemer screw stock. It can be drilled, tapped, threaded and otherwise machined with ordinary machine shop tools and practice.

From a corrosion-resisting standpoint, selenium does not impair in any way the resistance of stainless steel to the salt spray test. Moreover, it seems to increase the resistance of the metal to certain chemical solvents, such as boiling solutions of acetic acid or aluminum sulphate.

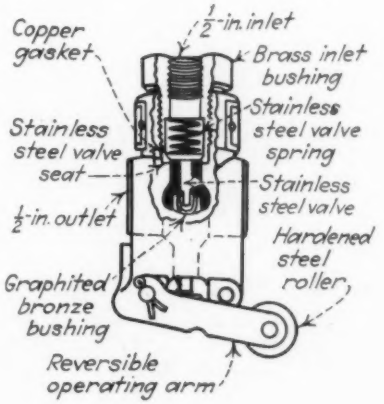
Allen-Bradley Bulletin 609 Starting Switch

A manually operated starting switch for small a.c. motors has been developed by the Allen-Bradley Co., 1311 S. First St., Milwaukee, Wis. This Bulletin 609 switch has two overload relays; tripping either one of them opens the contactor and completely disconnects the motor from the line. Either relay can be reset without opening the switch cabinet.

Size 1 has three poles; size 2 has three and four poles. The Size 1 switch will handle self-starting single-phase motors up to 1 hp., 110 volts, 2 hp., 220 volts, and polyphase motors up to 2 hp. at 220, 440, and 550 volts. The Size 2 switch is rated at 5 hp., 220 volts, and 7½ hp., 440 and 550 volts, for 2 and 3-phase motors.

Briggs & Stratton Air Saver

Designed originally for service within its own plant, a leakproof air valve known as the "Air Saver" is now being presented to the general market by the Briggs & Stratton Corp., Milwaukee, Wis. The device supplies automatic, intermittent control of compressed air blasts for ejecting work, keeping dies and tools clean, etc. As compared to a steady flow of compressed air, it is claimed



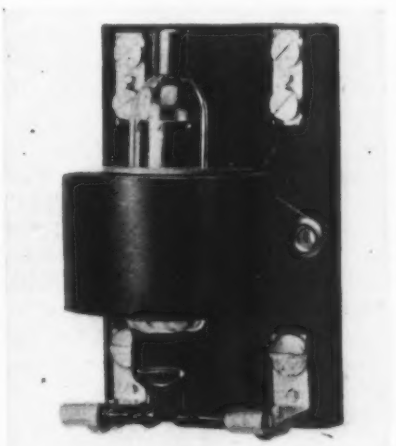
that the automatic, intermittent controlled blast, provided by the Air Saver eliminates waste of air, is easier on the operator, speeds up production and improves the quality of work. Valve, valve seat and valve spring are of stainless steel. The air inlet bushing is of brass. The long valve guide is lined with a special graphited bronze bushing. Reversible operating arm, roller and bearing pins are of hardened steel. In some installations several air jets are operated simultaneously, since the device has a large air capacity.

Outside dimensions are: height, 4½ in.; width, 3 in., these measurements being without mounting attachments. The device is supplied in one size only, which takes ½-in. standard pipe.

"Diamond H" Magnetrole Relay

This new development of the Hart Mfg. Co., Hartford, Conn., is made up of an especially designed glass tube inside of which there is a porcelain cup separating two concentric pools of mercury. Each pool is connected to terminals at the bottom of the base. Surrounding the porcelain cup is a metal cylinder having magnetic properties, so arranged that when moved it will displace the mercury in the outer pool, completing a mercury-to-mercury contact. The action of this cylinder is controlled by a solenoid or coil on the outside of the tube. The action is entirely magnetic.

The "Diamond H" magnetrole relay requires an operating current of 5 watts as a maximum for the 10 and 20 amp. sizes. There is no exposed arc, and the device will operate continuously up to 120 operations a minute. Mounting position is not critical, the tube working satisfactorily if tipped to an angle of 25 deg. It is particularly adapted for motor and light control, time and temperature control, signal equipment, photo-electric cell operation, and similar applications. Two capacities offered are 10 and 20 amp., 250 volts, but all voltages from 6 to 220 can be furnished.



New Books and Publications

"Endurance Limit and Design"

[*Dr. A. Thum and Dipl.-Ing. W. Buchmann.*] 82 pages, 6x7 in. Paper covers. Published by VDI-Verlag, Berlin, N.W.7., Germany, in the German language. Price 6.90 Marks.

Dealing entirely with the question of endurance limits, concentrated stresses, and the manner of making design calculations for these factors, this book is divided into four main parts. The first discusses the behavior of material under static loads and alternating loads, and the characteristics of fatigue fracture. The second portion presents the conception of endurance strength, discusses the method for determining the endurance limit, and the relation between endurance limit and ultimate strength, and describes testing methods and testing machines. The third chapter is devoted to a discussion of concentrated stresses resulting from notch effects such as encountered in abrupt changes in cross-section, and sets up methods for calculating the influences of such changes in section. In the final chapter are given rules for designing parts subject to alternating stresses and the method of applying the endurance-limit figures is explained.

"The Book of Metals"

[*Donald Wilhelm.*] 341 pages, 6x9¼ in. Bound in cloth boards. Indexed and illustrated. Published by Harper & Brothers, 49 E. 33rd St., New York, N. Y. Price $4.

This is the story of all the metals, their history, origin, mining, individual attributes, their principal alloys and their uses. Though written with the cooperation of metallurgists of the standing of Dr. Zay Jeffries, Dr. James Aston and others, and thus technically sound, it is easy to read.

Iron, wrought iron, cast iron, steel, steel alloys, aluminum, magnesium, beryllium, copper, brass, bronze, nickel, zinc, lead, tin, gold, silver and platinum are all treated authoratively, and with surprising completeness. As an additional reward for his reading, this reviewer was pleased to note that the author quoted George S. Brady, formerly managing editor of *Product Engineering*.

Colored Rubber Tires

[*The B. F. Goodrich Rubber Co., Akron, Ohio.*] "New Feathers for the Peacock" illustrates in color Goodrich Colorweld Silvertown tires as applied to automobiles of various color schemes.

Diesel Engines

[*Caterpillar Tractor Co., Peoria, Ill.*] A six-page folder entitled "Caterpillar D-9900 Engine" lists full specifications of the new diesel power unit described in *Product Engineering* for July, 1932.

Electric Equipment

[*General Electric Co., Schenectady, N. Y.*] Bulletin GEA-1586, entitled "Electric Equipment for Woodworking," supersedes 41521B and includes full data on electric motors and controls.

Electric Immersion Units

[*Harold E. Trent Co., 618-640 N. 54th St., Philadelphia, Pa.*] Leaflet TB-30, punched for insertion in a loose-leaf binder, describes the construction and application of the Trent Type IR immersion units and gives a list of sizes and capacities.

Everdur Metal

[*The American Brass Co., Waterbury, Conn.*] An eight-page illustrated bulletin has just been received describing the physical characteristics of Everdur metal with particular reference to its use as a high-strength material for welded rust-proof tanks.

Packaging

[*National Metal Edge Box Co., Callowhill at Twelfth St., Philadelphia, Pa.*] A portfolio entitled "The Improved Metal Edge Method of Packaging" presents full details.

Pumps

[*De Laval Steam Turbine Co., Trenton, N. J.*] Catalog B-5, entitled "DeLaval Single-Suction Multistage Pumps," describes a line built with as many as seven stages in one casing and for heads up to or exceeding 1,600 lb. per sq.in.

Research Bulletins

[*Department of Scientific & Industrial Research, 16 Old Queen St., Westminster, London, S.W.1, England*] Engineering Research Special Report No. 18 is the sixth of a series on the properties of materials at high temperatures. It gives the result of researches on the behavior of six heat-treated steels, namely 0.5 per cent carbon steel, 3 per cent nickel steel, nickel chromium steel, chrome vanadium steel, stainless steel, and Staybrite, plus three non-ferrous metals, namely phosphor bronze, duralumin, and 60-40 brass. Copies may be secured through the British Library of Information, 270 Madison Ave., New York, N. Y. Price 20 cents.

[*Engineering Experiment Station, University of Illinois, Urbana, Ill.*] Bulletin No. 240, "The Flow of Air Through Circular Orifices in Thin Plates" is the second of a series on the subject, the first having been Bulletin No. 270. Bulletin No. 241, "Strength of Light I-Beams" describes tests conducted to supply fundamental design data. Bulletin No. 242, "Bearing Value of Pivots for Scales," is probably the first technical report on the effect of the following variables: angle of pivot; width of edge; hardness of material.

Roller Bearings

[*Hyatt Roller Bearings Co. (Division of General Motors), Newark, N. J.*] Bulletin 100 bears the title "Dimensions and Load Rating Tables of Hyatt Roller Bearings & Their Application," and includes full data on wound roller-type bearings, solid roller-type bearings, single-row radial bearings, and lubrication data.

Splash-Proof Motors

[*General Electric Co., Schenectady, N. Y.*] Bulletin GEA-1619 describes splash-proof motors of the squirrel-cage induction type in horsepowers ½ to 75, standard speeds and voltages. These are designed particularly for applications in which open motor operation is handicapped by the presence of splashing or dripping liquids.

Squirrel-Cage Motors

[*Wagner Electric Corp., 6406 Plymouth Ave., St. Louis, Mo.*] Bulletin 174 describes multi-speed squirrel-cage motors in horsepower ratings of from ½ to 125. The description covers constant-torque, constant-horsepower and variable-torque motors further classified as two-speed, three-speed and four-speed motors, with illustrations of various applications.

Stainless Steel Sheets

[*American Sheet & Tin Plate Co., Frick Bldg., Pittsburgh, Pa.*] "American USS Stainless & Heat-Resisting Steel Sheets & Light Plates" presents physical characteristics and chemical properties of an unusually full line.

Wire Cloth

[*Newark Wire Cloth Co., Newark, N. J.*] General Catalog No. 32 gives full data on the selecting and testing of wire cloth in a wide range of meshes and materials.

Preferred Numbers

The accompanying table of Preferred Numbers was informally approved by the American Engineering Standards Committee and recommended to industry for a period of trial. It was published as Bulletin AESC-Z17—1927.

0.001 to 0.01			0.01 to 0.1				0.1 to 1.0				1 to 10				Values to 5 Figures	Term No.
5 Series	10 Series	20 Series	5 Series	10 Series	20 Series	40 Series	5 Series	10 Series	20 Series	40 Series	5 Series	10 Series	20 Series	40 Series		
											1	1	1	1	10592	1
														1.05	11220	2
													1.12	1.12	11885	3
														1.18	12589	4
												1.25	1.25	1.25	13335	5
														1.32	14125	6
													1.4	1.4	14962	7
														1.5	15849	8
											1.6	1.6	1.6	1.6	16788	9
														1.7	17783	10
													1.8	1.8	18837	11
												2.0	2.0	1.9	19953	12
														2.0	21135	13
													2.25	2.1	22387	14
														2.25	23714	15
											2.5	2.5	2.5	2.35	25119	16
														2.5	26608	17
													2.8	2.65	28184	18
														2.8	29654	19
													3.2	3.0	31623	20
												3.2		3.2	33497	21
														3.4	35482	22
													3.6	3.6	37584	23
											4.0	4.0	4.0	3.8	39811	24
														4.0	42170	25
													4.5	4.25	44668	26
														4.5	47315	27
														4.75	50119	28
												5.0	5.0	5.0	53089	29
														5.3	56234	30
													5.6	5.6	59566	31
											6.4	6.4	6.4	6.0	63096	32
														6.4	66834	33
													7.2	6.8	70795	34
														7.2	74990	35
												8.0	8.0	7.6	79433	36
														8.0	84140	37
													9.0	8.5	89125	38
														9.0	94405	39
											10.0	10.0	10.0	9.5	100000	40

Preferred Numbers are the rounded geometric series with the ratios:

$\sqrt[5]{10}$ (5 series); $\sqrt[10]{10}$ (10 series); $\sqrt[20]{10}$ (20 series); $\sqrt[40]{10}$ (40 series);

The exponents thus give the number of terms in one order of magnitude, (that is between 1 and 10, 10 and 100, etc.).

In all length measurements, the numbers apply to the inch as the unit of length. With all dimensions other than lengths, the numbers apply to the unit forming the basis of the standardization.

Preferred Numbers above 10 in. are formed by multiplying the numbers between 1 and 10 in. by 10, 100, etc. Preferred Numbers below 0.001 in. are formed correspondingly by division of the numbers between 0.001 in. and 0.01 in. by 10, 100, etc. Between 0.001 and 0.01 in. the 20 series will probably serve as the finest gradation; the 40 series is therefore not given.

So far as possible the numbers of the 5 series are to be used in preference to those of the 10 series, these again in preference to those of the 20 series, and these finally to those of the 40 series. It is permissible to pass over from one series of Preferred Numbers to an adjacent series.

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REFERENCE BOOK SHEET

Fractional Preferred Numbers

FRACTIONAL PREFERRED NUMBERS
1/16 to 1

* 5 Series	10 Series	20 Series	40 Series
60%	25%	12%	6%
1/16	1/16	1/16	
		9/128	
	1/8	1/8	
		11/128	
3/16	3/16	3/16	
		7/64	
	1/4	1/4	
		9/64	
5/16	5/16	5/16	
		11/64	
	3/8	3/8	3/8
			13/64
		7/32	7/32
			15/64
1/2	1/2	1/2	1/2
			17/64
		9/32	9/32
			19/64
	5/8	5/8	5/8
			21/64
		11/32	11/32
			23/64
3/4	3/4	3/4	3/4
			25/64
		13/32	13/32
			27/64
	7/8	7/8	7/8
			29/64
		15/32	15/32
			31/64
	1	1	1
1	1	1	1

* These are not exactly 5, 10, 20, and 40 series since the indicated number of steps are between 1/2 and 1.

† These are not exactly 1 1/3, 3 1/3, 6 2/3, and 13 1/3 series since the indicated number of steps are not between 0.1 and 1.

FRACTIONAL PREFERRED NUMBERS
1 to 100

5 Series	10 Series	20 Series	40 Series	5 Series	10 Series	20 Series	40 Series
60%	25%	12%	6%	60%	25%	12%	6%
1	1	1	1	10	10	10	10
			1 1/16				10 1/2
		1 1/8	1 1/8			11	11
			1 3/16				11 1/2
	1 1/4	1 1/4	1 1/4		12 1/2	12 1/2	12 1/2
			1 5/16				13 1/2
		1 3/8	1 3/8			14	14
			1 7/16				15
1 1/2	1 1/2	1 1/2	1 1/2	16	16	16	16
			1 9/16				17
		1 5/8	1 5/8			18	18
			1 11/16				19
	2	2	2		20	20	20
			2 1/16				21
		2 1/4	2 1/4			22	22
			2 3/16				23 1/2
2 1/2	2 1/2	2 1/2	2 1/2	25	25	25	25
			2 5/16				26 1/2
		2 3/4	2 3/4			28	28
			2 13/16				30
	3 1/4	3 1/4	3 1/4		32	32	32
			3 5/16				34
		3 1/2	3 1/2			36	36
			3 3/4				38
4	4	4	4	40	40	40	40
			4 1/4				42
		4 1/2	4 1/2			44	44
			4 3/4				47
	5	5	5		50	50	50
			5 1/4				53
		5 1/2	5 1/2			56	56
			5 3/4				60
6 1/4	6 1/4	6 1/4	6 1/4	64	64	64	64
			6 3/4				68
		7	7			72	72
			7 1/2				76
	8	8	8		80	80	80
			8 1/2				84
		9	9			88	88
			9 1/2				94
10	10	10	10	100	100	100	100

FRACTIONAL PREFERRED NUMBERS—SUPPLEMENTARY
1/16 to 100

† 1 1/3 Series	3 1/3 Series	6 2/3 Series	13 1/3 Series
300%	100%	40%	18%
1/16	1/16	1/16	
		11/128	
	1/8	1/8	
			11/64
		3/16	3/16
			13/64
1/4	1/4	1/4	1/4
			17/64
		5/16	5/16
			21/64
		3/8	3/8
			25/64
		7/16	7/16
			29/64
1	1	1	1
			1 1/16
		1 1/8	1 1/8
			1 3/16
	2	2	2
			2 1/16
		2 1/4	2 1/4
			2 5/16
		2 3/8	2 3/8
			2 7/16
4	4	4	4
			4 1/16
		4 1/2	4 1/2
			4 5/8
	8	8	8
			9 1/2
		11	11
			13 1/4
16	16	16	16
		22	22
			26 1/2
	32	32	32
			38
		44	44
			53
64	64	64	64
			76
		88	88

The fundamental Preferred Numbers shown on the preceding page are more or less international and therefore based on the use of decimals. In this country the use of fractions is so common that tables based on fractions for the range of 1/16 to 100 seem desirable. The use of these fractional numbers is made optional wherever it appears to be impractical to use decimals. Fractional numbers may be used for drawings and material stock, while decimals may be preferred for ratings, finished dimensions, etc., where the trade is now using decimals and is equipped with tools for measuring in decimals.

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